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NEW METHODS OF STUDYING NOISE AND VIBRATION AND CYBERNETIC DIAGNOSIS OF MACHINES AND MECHANISMS

Edited by K. M. Ragul'skis

Abstracts of papers from the All-Union Symposium held June 29 to July 1, 1970 at Kaunas Polytechnical Institute, Kaunas, 1970

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION • WASHINGTON, D. C. • MAY 1972

NEW METHODS OF STUDYING NOISE AND VIBRATION AND CYBERNETIC DIAGNOSIS OF MACHINES AND MECHANISMS

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Translation of "Novyye Metody Issledovaniy Shumov i Vibratsiy i Kiberneticheskaya Diagnostika Mashin i Mekhanizmov." Abstracts of papers from the All-Union Symposium held June 29 to July 1, 1970 at Kaunas Polytechnical Institute, Kaunas, 1970

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

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NEW METHODS OF STUDYING NOISE AND VIBRATION AND CYBERNETIC DIAGNOSIS OF MACHINES AND MECHANISMS

ABSTRACT. Brief summaries are presented of research work performed in the area of the study of noise and vibration and cybernetic diagnosis of machines and mechanisms. The reports were originally presented at an All-Union Symposium held June 29, 1970 through July 1, 1970.

OPTIMAL DETERMINATION OF CHANGES IN PROPERTIES OF RANDOM PHENOMENA

L. A. Tel'ksnys (Vil'nyus)

The problem is studied of determining the most probable moments in time for a change in the statistical properties of random phenomena on the basis of single or repeated realizations, fixed in observation intervals of finite length. /3*

This problem arises in many areas, in particular in the investigation of noise and vibration, as well as in the diagnosis of machines and mechanisms.

Statement of problem.

Let

$$X(t)=\begin{cases} X^{(1)}(t), & t\in(u_0,\ u) \\ X^{(2)}(t) & t\in(u,\ u_2), \end{cases}$$

be a random phenomenon with mathematical expectation

$$m(t)=\begin{cases} m^{(1)}(t), & t\in(u_0,\ u) \\ m^{(2)}(t), & t\in(u,\ u_2), \end{cases}$$

and with correlation function

$$k(\vartheta,\ \tau)=\begin{cases} k_1(\vartheta,\ \tau); & \vartheta,\ \tau\in(u_0,\ u) \\ k_2(\vartheta,\ \tau); & \vartheta,\ \tau\in(u,\ u_2). \end{cases}$$

$X^{(1)}(t)$, $X^{(2)}(t)$ are normal. Moment in time u, when random phenomenon $X(t), t\in(u,u_2)$ changes its statistical properties is unknown or random. The a priori distribution $\alpha(u)$ at instant u either is known, or it is assumed that it is constant over the interval in question, i.e., $\alpha(u)$ = const.

*Numbers in the margin indicate pagination in the foreign text.

It is required, using one or more realizations $x(t), t \in (u_0, u_2)$ of random phenomenon $X(t), t \in (u_0, u_2)$, to find the most probable estimate u* of parameter u. That is, we must determine the most probable moment in time u* when random phenomenon $X^{(1)}(t)$ will be converted to random phenomenon $X^{(2)}(t)$. /4

The problem is solved by determining the maximum of the a posteriori distribution density function of the desired parameter u. It is found on the basis of the maximum or minimum of the function, related in some manner to the a posteriori distribution density function [1,4].

Calculations for determination of u* can be performed in practice by computer.

Examples of solution of the problem by computer are presented.

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THE PROBLEM OF OPTIMAL SYNTHESIS OF DYNAMIC MODELS BY THE METHOD OF LP SEARCH

M. D. Genkin, V. K. Grinkevich, I. M., Sobol', R. B. Statnikov (Moscow)

The present work studies a new universal method for search for optimal parameters based on the use of Haar functions (2,4).

This method can be called LP search, since in place of the random points in a multidimensional cube, we use the point in an LP_τ sequence (2). These points are not random and are distributed much more evenly than random points. However, calculation of these points by computer is quite simple.

For further improvement of the parameters of the models, we used ordinary local methods (1). Incidentally, in many problems, no improvement was required, since a sufficiently good model was achieved in the first stage.

We note two more situations, in which the use of random, and particularly LP search, is quite expedient:

a) If it is necessary to analyze the possibility of optimization of a model on the basis of several different criteria or to study the influence of many parameters, this can be performed using the same test points;

b) If conditional conditions separate a nonconvex set in the space of models, convergence of local methods may occur only in a certain area around the optimum; this area can be reached by random or LP search.

We shall represent the inertial and rigidity parameters, as well as the damping coefficients, by $\alpha_1,\ldots,\alpha_r$. Permissible variations within the limits of the defined kinematic plan indicate that

$$\alpha_j^* \leqslant \alpha_j \leqslant \alpha_j^{**} \quad 1 \leqslant j \leqslant r \tag{1}$$

Then the dynamic model R_E[1] (with fixed kinematic structure E) is determined by the point $(\alpha_1,\ldots\alpha_r)$ in an r-dimensional parallelepiped (1).

[1]This formula for R_E corresponds to the following definition: models refer to two systems of objects A and B, between which a homomorphic mapping of system A in a certain system A' and a homomorphic mapping of B in a certain system B' can be established such that A' and B' are isomorphic. It is assumed that the relationship of isomorphism (homomorphism, isofunctionalism) obtains between the model and prototype (I).

In constructive realization of the model, the following problem arises: from the set of all R_E, select an optimal model which maximizes a certain criterial functional, dependent on the solution of the system, describing dynamic model R_E. In the final analysis, each such functional depends on the parameters $\alpha_1,\ldots,\alpha_r$. Therefore, we write it in the form of $\Phi_\gamma(\underline{\alpha})$ where the vector $\alpha = (\alpha_1,\ldots,\alpha_r)$, while γ can be called the quality index.

Assuming that in a certain closed area Γ belonging to parallelepiped (1)

$$\sup_{\alpha\in\Gamma} \Phi_\gamma = \Phi_\gamma(\alpha^+) \tag{2}$$

Then parameter $\underline{\alpha}^+$ will be referred to as the optimal parameter under condition Γ, while the corresponding model $R_E(\alpha^+)$ is referred to as the optimal model under condition Γ. For brevity, the values of $\Phi_\gamma(\underline{\alpha})$ where $\alpha\in\Gamma$ will be written in the form $\Phi_\gamma(\underline{\alpha}|\Gamma)$.

Using the plan presented in [2] (p. 219), the points $Q_l = (q_{l,1},\ldots q_{l,N})$, were calculated, forming the LP_τ sequence in the N-dimensional cube. For each point HQ_l, a test point $\underline{\alpha}^{(l)} = (\alpha_1^{(l)},\ldots,\alpha_N^{(l)})$ was determined in the parallelepiped [1]: /6

$$\alpha_j^{(l)} = \alpha_j^* + q_{l,j}(\alpha_j^{**} - \alpha_j^*) \tag{3}$$

Then the system of equations describing the dynamic object being studied with parameter $\underline{\alpha}^{(l)}$ was solved, and the quantities of interest were calculated. By looking through the sequence of points (l = 0, 1, 2,...,k - 1), the p "best" points were selected, from which "improvement" of the parameters was performed by regular methods. However, the value of the functionals studied was improved only slightly by regular methods in comparison with the best of the p points.

Thus, these studies confirmed the effectiveness of global LP search in comparison with random search (the number of statistical tests is significantly reduced and the probability of finding the global extreme for functionals with multiple extremes is increased), and in many cases improvements are made in comparison with any regular search methods.

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ESTIMATION OF DOMINANT PARAMETERS OF DYNAMIC SYSTEMS

I. I. Baltrunas (Vil'nyus)

Statement of Problem. The term "dominant parameters of a dynamic system" refers to: the transfer coefficient, time constant and pure delay.

The problem consists of constructing a self-tuning model, its investigation and estimation of the dominant parameters of the dynamic system.

Symbols. $x(t)$ is the input signal (it is assumed that $x(t)$ is a random process, stable in the broad sense); $z(t) = y(t) + n(t)$ is the observed signal at the output of the object; $y(t)$ is noise (assumed to be a stable Gaussian process similar to "white noise").

$\hat{y}(t) = \hat{A}(t)\phi(x(t),\hat{B}(t))$ is the output signal of the model, $\hat{A}(t)$ and $\hat{B}(t)$ are in the general case the vectors of the tuned parameters of the model at moment in time t; $\hat{A}(t)$ is related to $\hat{y}(t)$ linearly and to $\hat{B}(t)$ --nonlinearly; $\phi(x(t)\ \hat{B}(t))$ is a known vector function of its argument x, $\hat{B}$. /7

$\gamma_{\hat{A}}(t), \gamma_{\hat{B}}(t)$ are defined functions of time (in particular, constants); γ is the interval of gliding summation, T is the length of realizations of processes $x(t)$, $z(t)$.

The self-tuning algorithm is achieved using the following equations:

$$\frac{d}{dt}\hat{A}(t) = -\gamma_{\hat{A}}(t)\frac{\partial Q_l(\hat{A}_t, \hat{B}_t, t)}{\partial \hat{A}(t)},$$

$$\frac{d}{dt}\hat{B}(t) = -\gamma_{\hat{B}}(t)\frac{\partial Q_l(\hat{A}_t, \hat{B}_t, t)}{\partial \hat{A}(t)},$$

$$0 \leqslant t \leqslant T, \quad l = \overline{1,3}.$$

The quality criterion Q_l of the adaptive process consists of the following functionals:

$$Q_1(\hat{A}_t, \hat{B}_t, t) = \left[z(t) - \hat{A}(t)\,\varphi\left(x(t), \hat{B}(t)\right)\right]^2,$$

$$Q_2(\hat{A}_t, \hat{B}_t, t) = \frac{1}{\lambda}\int_{t-\lambda}^{t}\left[z(s) - \hat{A}(s)\,\varphi\left(x(s), \hat{B}(s)\right)\right]^2 ds, \quad \lambda > 0$$

$$Q_3(\hat{A}_t, \hat{B}_t, t) = \frac{1}{T}\int_{0}^{T}\left[z(t) - \hat{A}(t)\,\varphi\left(x(t), \hat{B}(t)\right)\right]^2 dt.$$

One characteristic feature of problems of this class is that the vector of parameters of $\hat{B}$ (for example, the time constant of the inertial length, delay) is included nonlinearly in the equation of the circuit.

The adaptive system is studied by the method of statistical modeling by digital computer, i.e., a discrete analog of the continuous system is studied. Investigations are performed for:

1. Determinability of the dominating parameters of the dynamic system as a function of the statistical characteristics of the input signal x(t).

2. The accuracy of the estimation of these parameters as a function of the type of adaptation algorithm and the intensity of noise n(t). For this purpose, the following algorithms are compared: the gradient algorithm, algorithm with gliding summation, stochastic approximation algorithm and the least squares algorithm. /8

3. Certain recommendations are presented on utilization of these algorithms.

ADAPTIVE MODEL FOR EVALUATION OF WEIGHT FUNCTION AND LEVEL OF NOISE OF LINEAR DYNAMIC SYSTEM

A. A. Nyamura and K. I. Rashchyus (Kaunas)

A study is presented of the process of adaptation in an adaptive model, designed for estimation of the weight function and noise level of a linear dynamic system. Since the weight function of the object is not accessible to direct observation, calculation of the weight coefficients of the model is performed on the basis of results of observation of perturbation of x(t) and the output coordinates of the object z(t) by gradient methods and stochastic approximation. Formulas are presented for determination of the speed of the adaptation process, i.e., the process of automatic adjustment of weight coefficients of the model in the case of a quasi-stable object and when noise n(t) at its output is white noise. It is demonstrated that the adaptation time is less when the adaptive system operates using the gradient algorithm. The duration of adaptation of a discrete adaptive system operating using the algorithm of steepest descent is studied.

An experimental study is presented of a discrete adaptive system by the method of statistical modeling using the UMI digital control machine. Investigation was performed with realization of three characteristic types of input perturbation of x_i with three levels of noise n_i. The dependence of the rate of tuning of the weight coefficients on the form of correlation function of the input signal and on the noise level at the output of the object is demonstrated.

STUDY OF STOCHASTIC APPROXIMATION ALGORITHM BY THE "MONTE CARLO" METHOD, USED TO ESTIMATE THE WEIGHT FUNCTION

N. A. Arbachyauskene (Kaunas)

The estimation of the weight function of a linear dynamic system by the method of stochastic approximation is analyzed for the case when the number of results of observations is finite. Estimates of the vector of weight coefficients B are calculated using the formula: /9

$$\hat{B}_{l+1}(i+1)=\hat{B}_{l+1}(i)+\gamma(i)\left[z(i+1)-F_{l+1}^{T}(i)\,\hat{B}_{l+1}(i)\right]F_{l+1}(i), \quad (1)$$

where $\hat{B}_{l+1}$ is an estimate of the vector weight coefficient B $(b_0,b_1,\ldots,b_l)^T$; ff(i) is the input signal; $F_{l+1}(i) = (f(i),f(i-1)\ldots f(i-l))^T$ is the vector of the input signal; zz(i) is the output signal observed with noise, i.e., $z(i) = x(i) + n(i)$; n(i) is the noise at the output of the object; $\gamma(i)$ is a series of positive numbers, satisfying the condition $\sum_{i=0}^{\infty}\gamma(i) \to \infty$; $\sum_{i=0}^{\infty}\gamma^2(i) < \infty$, i is the iteration number, l is the number of components of vector B and F.

A sufficiently precise estimate of vector $\hat{B}_{l+1}$ by the method of stochastic approximation, particularly with a high noise level, can be produced only after a tremendous number of iterations (10^4-10^5). The determination of this great number of points of realizations of input and output signals requires a great deal of time. Therefore, the possibility is studied of using a repeated listing algorithm, the theoretical basis of which was provided by B. M. Litvakov.

The accuracy of restoration of a parameter in a limited number of iterations depends on the statistical characteristics of the input signal f(i), noise level n(i) and the selection of a tuning step length $\gamma(i)$. Only parameter $\gamma(i)$ can be changed freely.

Investigation by the Monte Carlo method allowed certain specifics of the algorithm with repeated listing to be established. The influence of the following factors on the accuracy of estimation was determined: a) Input signal correlation function; b) noise at the output of the object; c) number of re- /10 results of observations. Practical recommendations are given for selection of the tuning step length T(i) as a function of these factors.

ESTIMATION OF PARAMETERS OF NONLINEAR HAMMERSTEIN OPERATOR

G. A. Rubikas (Kaunas)

A nonlinear, univariate, inertial object of known structure is identified by a Hammerstein model of the following form:

$$l_k = y_k + u_k = \sum_{j=0}^{m} \sum_{i=0}^{n} v_j c_i \varphi(x_{k-j}) + u_k, \quad k = 1, 2, \ldots \tag{1}$$

here I_k is the measured output of object y_k together with additive noise u_k acting on the output; $\{\phi_i(x)\}$ is a system of linearly independent functions of input signal x, ν_j;

$j = \overline{0,m}$, $i = \overline{0,n}$ are unknown coefficients, estimating the corresponding parameters of the objects.

Various mathematical methods allow us to create various algorithms for determination of ν_j, c_i. In order to improve the iteration procedure, we replace the paired products $\nu_j c_i$ by a single coefficient h_{ji}. Calculation can be performed using the following two algorithms: 1) an algorithm produced by the least squares method (the MNK algorithm) and 2) an algorithm produced by the stochastic approximation method (the SA algorithm).

The MNK algorithm:

$$h_{\nu+1} = \Gamma_{\nu+1}^{-1} L_{\nu+1}$$

here $h_{\nu+1}$ is the vector $\hat{h}$ in the $(\nu + 1)$-th iteration,

$\Gamma_{\nu+1}^{-1}$ is a matrix considering the input signal x in the $(\nu + 1)$-th iteration

$$\Gamma_{\nu+1}^{-1} = \left(1 + \frac{1}{\nu}\right) \Gamma_\nu^{-1} - (\nu + \varphi'_{\nu+1} \Gamma_\nu^{-1} \varphi_{\nu+1})^{-1} \frac{1}{\nu^2} \Gamma_\nu^{-1} \Phi_{\nu+1} \Gamma_\nu^{-1}$$

where

$$\Phi_\nu = \varphi_\nu \varphi'_\nu,$$

when /11

$$\varphi'_\nu = \Big[\varphi_\nu(x_{\nu-0}), \ \ldots, \ \varphi_n(x_{\nu-0}), \ \varphi_0(x_{\nu-1}), \ \ldots,$$

$$\varphi_n(x_{\nu-1}), \ \ldots, \ \varphi_0(x_{\nu-m}), \ \ldots, \ \varphi_n(x_{\nu-m})\Big];$$

$$L_{\nu+1} = \frac{\nu}{\nu+1} L_\nu + \frac{1}{\nu+1} l_{\nu+1} \varphi_{\nu+1},$$

here $l_{\nu+1}$ is the output of the object measured together with noise $u_{\nu+1}$, $L_0 = l_0 \Gamma_J, \phi_0^{-1} = aI$, where I is a unit matrix;

a is a certain number, $1 \leq u \leq 1000$.

The SA algorithm:

$$h_{\zeta\eta}^{\nu+1} = h_{\zeta\eta}^{\nu} + \frac{1}{\nu^{\alpha} \sum_{j=0}^{m} \sum_{l=0}^{n} \varphi_j^2 (x_{\nu-j})} \left[l_{\nu+1} - \sum_{j=0}^{m} \sum_{l=1}^{n} h_{jl}^{\nu} \varphi_l (x_{\nu+1-l}) \right] \varphi_\zeta (x_{\nu+1-\eta})$$

here $h_{\zeta\eta}^{\nu+1}$, $\eta = 0, m$, $\xi, = 0, n$ is the $\xi\eta$-th coefficient in the $(\nu + 1)$ the iteration

$\{ \phi_i; (x) \}$ is a system of linearly independent functions of the input signal x;

ν is the iteration number, $0 \leqslant \alpha \leqslant 1$ $(\alpha = 0.55)$.

The algorithms suggested allow the coefficients of expression (1) to be determined with a significant level of noise at the output. It should be noted that the iteration speed and quality of the MNK algorithm is better than of the SA algorithm.

ESTIMATION OF PARAMETERS AND CHARACTERISTICS OF NONLINEAR DYNAMIC SYSTEMS

A. K. Nenorta (Kaunas)

Certain methods are studied for determining the characteristics and parameters of nonlinear dynamic systems on the basis of observations of their input quantities x(t) and output quantities y(t).

A nonlinear dynamic system can be described approximately by a segment of the functional Volterra series /12

$$\hat{y}(t) = w_0(t) + \int_0^\infty w_1(\lambda_1)\, x(t-\lambda_1)\, d\lambda_1 + \\ + \int_0^\infty \int_0^\infty w_2(\lambda_1, \lambda_2)\, x(t-\lambda_1)\, x(t-\lambda_2)\, d\lambda_1\, d\lambda,$$

where $w_0(t)$ is a function, dependent on the initial state of the system;

$w_1(\lambda_1)$ is a weight function, defining the linear portion of the output quantity;

$w_2(\lambda_1,\lambda_2)$ is a weight function, defining the second order component of the output quantity;

$\hat{y}(t)$ is the output value of a model of the dynamic system.

In order to estimate the nonlinear dynamic system operator being studied, we assume $w_0(t) = 0$. Estimates will be sought in the following form:

$$\hat{w}_1(\lambda_1) = \sum_{j=1}^{k} a_{j0}\, \varphi_j(\lambda_1), \tag{2}$$

$$\hat{w}_2(\lambda_1, \lambda_2) = \sum_{j=1}^{k} \sum_{p=1}^{k} a_{jp}\, \varphi_j(\lambda_1)\, \varphi_p(\lambda_2), \quad (j \geqslant p), \tag{3}$$

where $\phi_j(\lambda_1), \phi(\lambda_2)$ is a system of known continuous functions, satisfying the condition $\phi_j(\lambda_1) = \phi_p(\lambda_2)$, if $j = p$ and $\lambda_1 = \lambda_2$, the method of selection of which is discussed in this report;

a_{jp} ($j = 1,2,\ldots,k$; $p = 0,1,\ldots,k$) are unknown coefficients, estimates of which must be found. It is assumed that the weight functions $w_1(\lambda_1)$ and $w_2(\lambda_1,\lambda_2)$ can be precisely approximated by functions $\hat{w}(\lambda_1)$ and $\hat{w}_2(\lambda_2)$ respectively. Considering (2) and (3), expression (1) can be written as follows:

$$\hat{y}(t) = \sum_{j=1}^{k} a_{j0}\, x_j(t) + \sum_{j=1}^{k} \sum_{p=1}^{k} a_{jp}\, x_j(t)\, x_p(t), \quad (j \geqslant p). \tag{4}$$

where we represent

$$x_\eta(t)=\int_0^\infty \varphi_\eta(\lambda)\,x(t-\lambda)\,d\lambda,\quad (\eta=1,\ 2,\ \ldots,\ k) \tag{5}$$

If in the process of production of data, all values of the input and output quantities x(t) and y(t) are precisely defined, calculation of estimates of coefficients a_{jp} requires as many values of quantities x(t) and y(t) as are required to determine estimates of coefficients a_{jp}. But in practice we can measure precisely only input quantity x(t). The measurement error n(t) in output quantity y(t) will be considered additive noise, normally distributed. Then estimate $\hat{a}_{jp}$ can be sought by solving the redundant system of equations /13

$$z_i-\Big[\sum_{j=1} \hat{a}_{j0}\,x_j(i)+\sum_{j=1}\sum_{p=1} \hat{a}_{jp}\,x_j(i)\,x_p(i)\Big]=0,\quad (j\geqslant p;\ i=1,\ 2,\ \ldots.) \tag{6}$$

by the method of least squares or the method of stochastic approximation, where z(i) = y(i) + n(i) is the result of observation of value y(t) in the presence of noise n(t) at discrete moments in time t_s, s = 1,2,...

The accuracy of the estimates produced was determined by experimental studies of these methods using the BESM digital computer by the method of statistical testing.

IDENTIFICATION AS A TASK OF SEQUENTIAL STATISTICAL ANALYSIS

A. A. Nyamura (Kaunas)

For the problem of identification, it is necessary first of all to differentiate classes with the following characteristics: nonlinear or linear, stable or unstable, inertial or noninertial. After a sufficiently narrow class of an operator has been defined, the operator itself becomes known with an accuracy to a certain vector parameter c, i.e., we have the following functional dependence:

$$y(t)=A[u(t)]=\Phi[u(s),\ 0\leqslant s\leqslant t;\ c(s),\ 0\leqslant s\leqslant t;\ t],$$

where A[·] is the mathematical operator of the system, unknown in advance, requiring determination; u(t) and y(t) are the input and output generalized vector coordinates of the system, which may be deterministic or random functions of scalar argument T. The last t in the formula indicates that the functional dependence Φ[·] may change as time passes. Variable u(t) and y(t) are observed /14 under the influence of random noise. It then remains to estimate the unknown vector parameter c.

Criteria of optimality of plans of observation and some problems of recognition of the class of operator A are studied, and a complete analysis of methods of estimation of vector parameter c is performed.

DIFFERENTIATION OF RANDOM FORCED OSCILLATIONS AND SELF-OSCILLATIONS PERTURBED BY RANDOM ACTIONS

A. A. Gorbunov and M. F. Dimentberg (Moscow)

The problem consists in determining whether a measured narrow band random process at the output of an oscillating system consists of forced oscillations of the system under the influence of external wide band random perturbations or whether the system is in the state of self-oscillations, modeled by random noise as a result of random perturbations at the input of the system. The solution to this problem in many cases can indicate what source of excitation is making the basic contribution to the vibration of the mechanical system.

Two recognition criteria are suggested for an object which can be represented by a second order equivalent quasi-linear system, based on analysis of the probability densities of the envelope of the output process or the process itself. The first criterion follows from solution of the Fokker-Planck equation, corresponding to an abbreviated equation for the amplitude. The second criterion is produced on the basis of the known integral dependence between the probability densities of a narrow band random process and its envelope. The accuracy of the second criterion has been tested by analog computer modeling and was found to be good even in the case of rather short intervals of observation of the output process.

DETERMINATION OF PARTIAL DAMPING COEFFICIENTS OF LINEAR SYSTEMS FROM MEAN PERIODS OF ENVELOPES OF RANDOM OSCILLATIONS

M. F. Dimentberg and A. R. Abul'khanov (Moscow)

A formula is derived for calculation of partial damping coefficients of linear systems on the basis of values of the resonant frequency and mean number of intersections of the envelope of a narrow band random oscillation with its mathematical expectation. The formula is concluded in the consideration of the influence of the filter-analyzer used to separate the oscillations in the degree of freedom being studied. Results are presented from experimental checks of the accuracy of the method using electrical models. /15

INTEGRAL EQUATIONS AS A METHOD OF DETERMINING THE SPECTRUM OF NORMAL MODES OF BACKGROUND VIBRATIONS

P. F. Ovchinnikov (Odessa)

Methods are studied for determining the spectrum of natural oscillations of systems, the oscillations of which are described by linear, loaded integral equations with an essentially positive kernel and a non-monotonic distribution function. In contrast to a spectrum with a monotonic distribution function, the natural spectrum of these equations falls on both sides of the number axis. It is demonstrated that neither methods of composicion of frequency equations and their conversion to secular form, nor methods of expansion of the secular equation to a polynomial can be extended to equations with non-monotonic distribution functions.

The possibility of using the method of successive approximation ior calculation of the natural number with the lowest absolute value is demonstrated.

Extension of the method of wakes allows any natural number to be defined as to absolute value. After demonstrating that the problem of determination of natural numbers can be reduced to a certain variational problem, the possibility of using the method of Ritz is established.

APPLICATION OF CORRELATION ANALYSIS FOR DIAGNOSIS OF THE CONDITION OF MECHANISMS

I. V. Alekseyev and E. P. Pyshkina (Moscow)

Knowledge of such oscillating characteristics of parts of mechanisms as the damping factor, natural frequency and spectral noise power density is of interest for purposes of acoustical diagnosis. /16

In order to determine these characteristics and estimate the influence of design parameters of parts on the process of noise formation, the Department of Engines of the Moscow Highway Institute has produced a motorless modeling installation, a block diagram of which is shown on Figure 1. This installation, using a correlator produced in Department E-9 of the Bauman Moscow Higher Technical School, has been used to determine the characteristics of individual parts mentioned above.

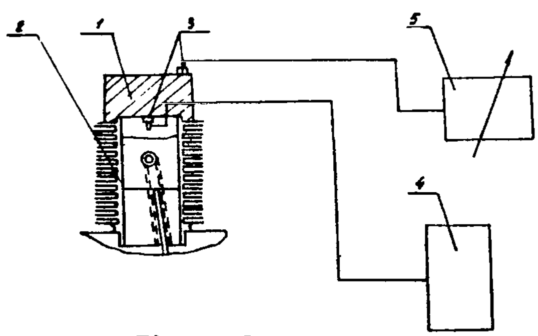

Figure 1.

The parts tested were models of cylinder heads of internal combustion engines. Four versions of heads were studied, differing from each other in thickness of end wall and construction material. The cylinder heads have the following radiating wall thicknesses: 3 ± 0.05 mm; 4.5 ± 0.05 mm; 5.25 ± 0.05 mm; and 6 ± 0.05 mm, and were made of type 45 and 18Kh steels. The selection of a design for a thin wall head was based on the following considerations:

a) The end wall of the head is made so that its rigidity is much less than the rigidity of the remaining elements of the design of the installation, allowing the assumption that the primary share of acoustical energy radiated by /17 the cover is determined by oscillations of its thin wall portion;

b) The design of the head allows analytic calculation of the natural frequency of its radiating surface, making it possible to evaluate the reliability of the experimental data.

Experimental determination of the damping factor and natural frequency of heads was performed on the basis of the parameters of the correlation function of the random process, measured using a vibration sensor installed on one side of the radiating surface of the head. The vibrator was installed symmetrically to the vibration sensor on the other side of the head, allowing the form of excitation during the experiment to approximate the form of actual exciting forces acting on the radiating element and thus avoiding undesirable asymmetrical oscillations and their corresponding natural frequencies.

Voltage from a noise generator was sent to the vibrator. The voltage from the vibration sensor was sent to a correlator, which was used to determine the parameters of the correlation function. The oscillating characteristics of the head are unambiguously related to the parameters of the correlation function. The analysis of theoretical and experimental data for determination of these characteristics showed good correspondence. The results produced demonstrate that the use of a correlation analyzer significantly facilitates the measurements performed.

The use of correlation analysis hardly excludes the use of classical methods of acoustical measurements. The expediency of the use of any given method is determined by the practical situation: availability of apparatus, its complexity, cost, nature of the process being studied, etc.

METHOD AND DEVICE FOR QUASI-PARALLEL ANALYSIS OF SHORT-TERM ACOUSTICAL SIGNALS

I. V. Alekseyev and V. N. Lukanin (Moscow)

We have developed a measurement channel for spectral analysis of noise and vibrations of engines during operation (Figure 1). It consists of a set of 24 one-third octave filters, the inputs of which receive the signal to be analyzed from the vibration sensor installed on the engine and first subjected to amplification. A switching device, controlled by pulses marking the /18 top dead center point, connects the outputs of each of the filters in turn to the recording device, an 8-beam cathode ray oscilloscope with a photographic attachment. The switching device used is a stepping switch, the contacts of which are closed in pairs. This connection of contacts allows the signal from each filter to be fed to the oscilloscope during two full crank shaft rotations, which corresponds to the length of the operating cycle in a four stroke engine. Thus, the signal being studied is recorded after having been passed through the one-third octave filters.

The coil of the stepping switch is connected to the anode circuit of a thyratron, which is opened by a pulse. Recordings of noise, vibrations, as well as the processes defining the operation of the engine, the indicator diagram and travel of the sprayer needle, performed using this channel, demonstrate the possibility of its utilization for diagnostic purposes. The pulses of vibration and noise resulting from the impact interactions in the structures of the engine are separated and can be analyzed to find relationships with parameters characterizing the technical condition of the engine.

USE OF A CORRELATOR BASED ON ORTHOGONAL FILTERS IN ACOUSTICAL MACHINE DIAGNOSIS

Yu. I. Bobrovnitskiy, M. D. Genkin and K. D. Morozov (Moscow)

It is suggested that the internal parameters of machines (diagnosis parameters) be measured using an orthogonal filter correlator. It is demonstrated that the parameters of the filter at the input of one of the channels can be selected so that a signal is produced at the output of the correlator which is proportional to the measured machine diagnosis parameter. The construction of such a filter requires that certain correlation measurements be performed in advance with several known values of the diagnosis parameter, i.e., that a "learning process" is needed.

A model diagnostic experiment is performed. The model of the diagnostic signal was a mixture of two random, strongly correlated signals, the amplitude of one of which was proportional to the measured diagnosis parameter. After learning, the correlator measured this parameter with an error never exceeding 10%.

DETERMINATION OF EQUIVALENT RANDOM PERTURBATION DURING MODELING OF COMPLEX DYNAMIC SYSTEMS

L. A. Manashkin and N. G. Baranov (Dnepropetrovsk)

Optimization of the parameters of damping devices requires that a model of /19 the equipment be perturbed so that the damping devices operate under near normal conditions. This is done by feeding a voltage from a noise generator with a rather broad spectrum to a synthesizer. The signal formed by the synthesizer is fed to an electrical model. The output quantities of the model are formed so that they can be compared with the results of experimental studies from a (standard) actual device under acutal conditions. The results of investigation of oscillations of the link of interest in the actual device are represented as the spectral characteristics of these oscillations. The electrical voltage produced in the model and corresponding to the oscillations of the unit being studied is sent to an analyzer. The results of analysis are compared with the fixed spectrum produced preliminarily during experimental studies of the oscillations of the actual equipment. By controlling the synthesizer, approximate correspondence of the spectra produced experimentally and by modelling can be achieved. The corresponding perturbation will be equivalent to actual perturbation from the standpoint of the operating conditions of the unit being studied.

A block diagram of the system as a whole, a block diagram of the synthesizer and of the analyzer are developed.

IDENTIFICATION OF MODEL OF HUMAN OPERATOR WITH RANDOM VIBRATION ACTIONS

L. V. Goykhaman, B. A. Potemkin and K. V. Frolov (Moscow)

A dynamic human operator system influenced by wide band random (white noise) perturbation in the frequency range up to 200 Hz is studied.

The problem is solved of determining the parameters of the dynamic human operator system and construction of a model for each position. The body of the operator is looked upon as a linear dynamic model consisting of series-connected absolutely rigid bodies with masses m_i, connected by elastic and damping elements k_i and C_i. /20

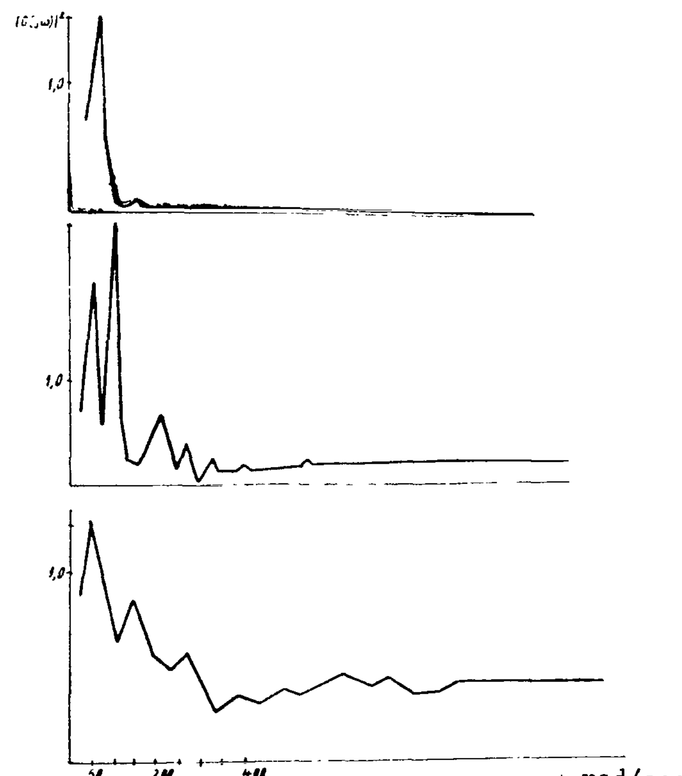

Figure 1.

A method is presented for selecting the length of the recording of the input process to assure its stability. Distribution rules of the output process are found for each position, and the correlation functions and spectral density functions are calculated for the input and output processes.

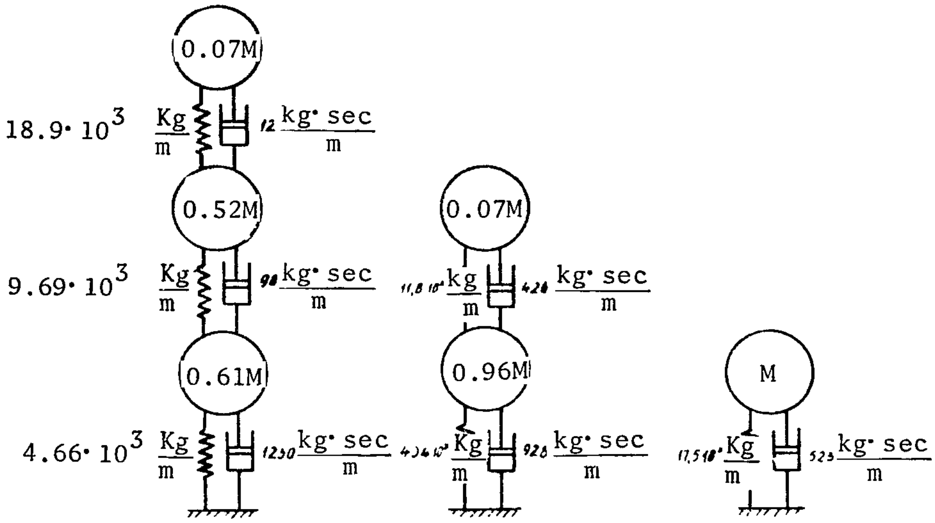

Figure 2.

The number of equations of motion of the model is determined by the form of the frequency characteristic for each position. The equations studied were /21 used to find analytic expressions for the amplitude-frequency responses:

$$G(\omega) = \frac{\sum_{i=0}^{n} B_i \omega^i}{\sum_{i=0}^{m} A_i \omega^i} \qquad (1)$$

The area limitations are fixed in the form $\Sigma m_i = M$, where M is the mass of the human operator.

By selecting N values of the experimental frequency response at frequencies ω_j and setting them equal to expression (1) for each of the frequencies, a system of nonlinear algebraic equations is produced relative to the model parameters to be determined.

In this case, the nodal frequencies correspond to the extreme points of the experimental frequency characteristic. Calculation of the model parameters m_i, c_i and k_i was done on the BESM-3M computer.

The possibility is demonstrated of using an algorithm allowing calculation of parameters by this method with a fixed accuracy. In this case, the /22 number of nodal frequencies N may exceed the number of unknown parameters.

As a result of the work, dynamic models are produced for each of the positions with numerical values of the parameters.

MULTICHANNEL DIAGNOSIS OF MACHINES BY DIGITAL COMPUTER

B. D. Tartakovskiy, A. I. Vyalyshev and B. A. Kanayev (Moscow)

Existing methods of diagnosis of machines and mechanisms based on measurement of their vibration-acoustical characteristics have the significant shortcoming that the characteristics of the acoustical elements of structures of machines and the bases on which they rest, as well as sectors of the sound conductors along which vibrations and acoustical waves propagate for reception by the test instruments, influence the recorded results. This influence, being essentially dependent on the point of placement of the test vibration and sound sensors, in most cases cannot be theoretically calculated. However, experimental checks of this influence have shown that it is practically impossible to eliminate it by proper selection of the point.

One method of suppressing this influence, which must be analyzed for most cases of realization of diagnosis systems as a random category, consists in using multichannel acoustical diagnosis systems (MSD). An experimental study was performed for selection of MSD parameters, consisting in the determination of expedient frequency bands and distances between sensors.

The machine selected for study was a defective electric motor, installed on a long, rigid frame. The sensors used were piezoelectric vibrometers, placed on the frame at various distances from the motor. The measurements performed in the stable mode utilized filters of various types with various widths of frequency band. The results were processed by digital computer using a special algorithm based on statistical methods.

Values characterizing the average change in vibrations resulting from the appearance of a defect were found. The minimum band widths of filters and minimum distances between vibration receptors providing the opportunity to /23
analyze the signal values produced at the certain points on certain frequency bands as noncoherent were determined.

This allowed the maximum number of independent values of vibration-acoustical signals occurring in the experiment to be determined, and on this basis allowed the confidence intervals for statistical estimation of the reliability of the mean difference between signals related to a defect to be found, i.e., allowed estimation of the reliability of the diagnostic effect. In the experiment performed, the random change in the mean signal level resulting from noise at individual points was only slightly less than the difference between levels resulting from the defect. However, the use of this method of data processing for data produced using multichannel measurements allowed the existence of a defect in the electric motor to be established with a probability of no less than 0.8.

CYBERNETIC METHOD OF STUDYING VIBRATION ACOUSTICAL CHARACTERISTICS OF COMPLEX STRUCTURES

B. D. Tartakovskiy and G. S. Lyubashevskiy (Moscow)

The essence of an algorithm which minimizes a set of measurements [1] consists of calculation of the most suitable increment for the argument (from a finite series of discrete values accepted) for the next single measurement based on information already accumulated in digital computer storage concerning behavior of the relation being studied. As a result, the density of points of measurement becomes uneven: it increases near sectors of high information value--extremes--and decreases between them.

The effectiveness of the algorithm with variable step in the change of the argument depends, of course, on the class of dependence being studied. Machine modeling of an algorithm as applicable to the frequency response of an oscillating link with one degree of freedom in the audio frequency range with optimally selected control parameters included in the algorithm indicated that

1) Reduction of the measurement time (number of measurements) using this algorithm in comparison with equidistant division of the argument with equal accuracy of the search for the extreme is

$$B \approx \sqrt[3]{2^{2\bar{n}}}\left(10+\frac{16}{\varepsilon}\right)^{-1}\sqrt[5]{\Delta Q^3},$$

where n is the number of terms in the series of discrete values of the increment of the argument, Q is the figure of merit of the object being studied, $\Delta \leq 10^2$ is the ratio of the extreme value of the function to the zone of insensitivity, below which values of the function are not of interest and are not used in calculation, $1 \leq \varepsilon \leq \infty$ is the quality parameter included in the algorithm; /24

2) The reduction in measurement time using the algorithm suggested in comparison to a continuous linear change in frequency with conditions of equal accuracy of search for the extreme value of the function is

$$B \approx -\frac{\Delta}{32+\frac{51}{\varepsilon}} \cdot \frac{F}{\Delta f \cdot \sqrt[3]{Q}} \cdot \frac{1}{\sqrt[3]{\delta \ln \delta}},$$

where Δf is the width of the transmission band of the analyzing channel, F is the range of change of frequency, δ is the fixed error in search for the extreme. Furthermore, when reproducing functions according to the algorithm suggested, the displacement of the extreme by frequency and its expansion, arising with a continuous change in frequency, are eliminated in principle.

Thus, for example, where $\varepsilon = \infty$, $\Delta f = 1/3$, $Q = 20$, $F = 2 \cdot 10^4$, $\delta = 5 \cdot 10^{-2}$, $\Delta = 10$, the gain in time $B \approx 10^4$, which indicates the high effectiveness of the algorithm, particularly for a high-Q system with a narrow transmission band in the analyzing channel. As the number of extremal maxima q is increased, the effectiveness of the algorithm decreases approximately in the ratio of $2q + 1$.

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AUTOMATION OF PROCESSING OF EXPERIMENTAL DATA AND DEVELOPMENT RECOMMENDATIONS FOR SELECTION OF OPTIMAL VIBRATION-ABSORBING COATING

B. D. Tartakovskiy and B. A. Kanayev (Moscow)

Automation of the selection of recommendations designed for improvement of the vibration-acoustical properties of an object by vibration absorption on the basis of data from experimental studies of vibrations of the object /25 assumes knowledge of the dependence of the quantities characterizing vibration (mean amplitude, mean square amplitude, etc.) on the parameters of the vibration-absorbing coatings (VC) to be used. For example, assuming in the first approximation

$$(VC)_1 = \frac{\langle A_0^2 \rangle}{\langle A_1^2 \rangle} = \frac{\eta_1 + \eta_0}{\eta_0} = 1 + \frac{\eta_1}{\eta_0}$$

($\langle A_0^2 \rangle$ and $\langle A_1^2 \rangle$ are the mean square amplitude of oscillations of the object before and after use of the coating, η_0 is the loss factor of the untreated object, η_1 is the loss factor created by the coating), we can calculate the parameters of the coating providing a fixed decrease in mean square vibration amplitude with minimum weight.

For rods and plates when excitation is by a monochromatic signal of frequency ω, the mean square vibration amplitude can be represented as

$$\langle A^2 \rangle = \sum_{k=1}^{\infty} c_k z_k,$$

$$Z_k = \frac{1}{m\,[(p_k - \omega^2)^2 + \eta^2 p_k^4]}, \qquad p_k = \lambda_k \left(\frac{B}{m}\right)^{\frac{1}{2}},$$

where m is the specific mass, B is the bending rigidity, η is the loss factor, λ_k is a coefficient corresponding to the K-th root of the characteristic equation. It follows from the expression for Z_k that if $P_k \cong \omega$, then $z_K \sim 1/\eta^2 B^2$; where $\eta \ll 1$ and $p_k = \omega$, the value of $\langle A^2 \rangle \sim 1/\eta^2 B^2$, from which

$$(VC)_2 = \frac{\langle A_0^2 \rangle}{\langle A_1^2 \rangle} = \left(\frac{B_1}{B_0}\right)\left(1 + \frac{\eta_1}{\eta_0}\right)^2$$

where B_0 and B_1 are the bending rigidity of the structure before and after treatment.

Let us now assume that the spectrum of the modes of the structure excited by the oscillating forces is very broad, and in the first approximation we can consider that the amplitudes of the modes excited are approximately identical.

In this case we can limit ourselves to the power approximation, considering that the change in kinetic energy of the structure is inversely proportional to the loss factor. We then obtain

$$(VC)_3 = \frac{\langle A_0^2 \rangle}{\langle A_1^2 \rangle} = \left(\frac{B_1}{B_0}\right)^2 \left(1 + \frac{\eta_1}{\eta_0}\right).$$

The loss factor and bending rigidity of a two layer coating depend on the ratio of layer thicknesses of the coating [1,2]. The dependence of $(VC)_3$ on ratio H_2/H_1 of the thickness of the intermediate layer in a three layer structure to the thickness of the plate for various values of relative coating weight is shown on Figure 1. The nature of the change in frequency characteristics of $(VC)_2$ and $(VC)_3$ as a function of thickness relationship in a two layer coating is shown on Figure 2. Knowledge of these regularities allows us, on the basis of processing of experimental data, to perform computer calculation of the coating parameters most fully meeting the requirements for reduction in vibration level of the structure being measured. /26 /27

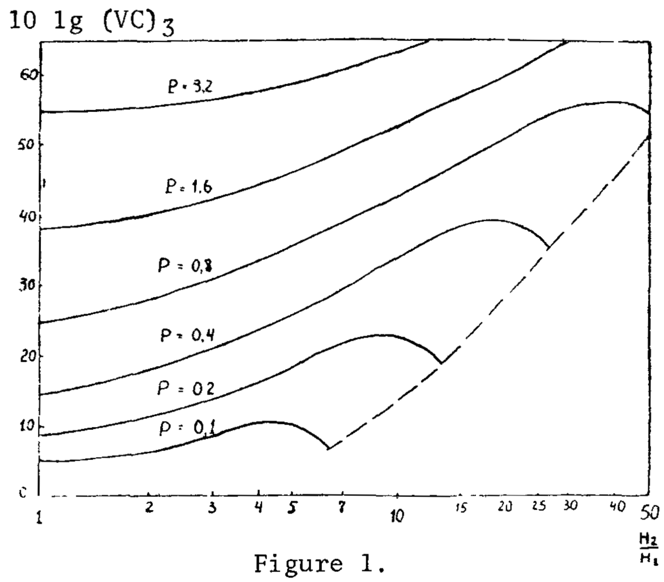

Figure 1.

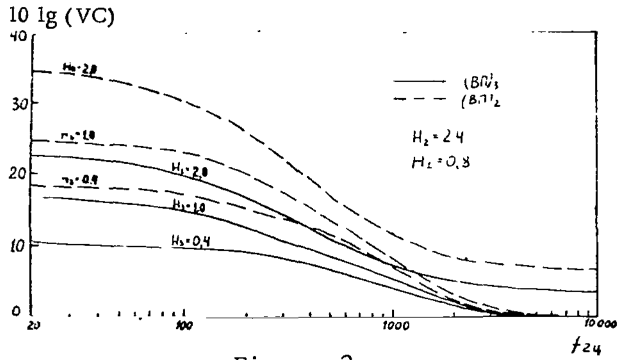

Figure 2.

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RANDOM OSCILLATIONS AND IDENTIFICATION OF PARAMETERS OF PRECISION BALL BEARINGS

A-E. Yu. Vitkute (Kaunas)

The reasons for vibration of ball bearings are errors in the shape of the rolling surface consisting of so-called waviness, changes in loads, changes in thickness of lubricant layer, which cannot be described as deterministic functions. Therefore, calculation and study of the corresponding dynamic system requires that probabilistic or statistical methods be used.

Within the framework of the correlation theory of functions, the degree of influence of individual factors on the total vibration level can be determined by solving an identification problem, in which the optimal estimate of the operator of a system in a class of linear operators is sought on the basis of the reactions of inputs and the output.

Determination of the distribution of system variables as the full characteristic of system behavior at each moment in time can be performed by statistical testing. Analytic study of the mathematical model is replaced by experimentation with a physical model, reproduced using a digital computer.

The principle stages in the solution are: a) determination of probabilistic models of actual random processes; b) generation of a sequence of random processes corresponding in their probability characteristics to actual processes in the system being studied; c) conversion of the sequence in correspondence to the structure of the actual system; d) statistical processing of the output signals produced. /28

During investigation, all characteristics of the bearing are measured in the washed state and after lubrication. Deviations from theoretical circularity of the tracks of the internal and external bearing rings are recorded using Indiron.

The actual processes are replaced with lattice functions. The recording interval and resultant sample space are determined according to the requirements for unbiased, consistent and optimal estimates in the sense of the minimum variance of estimates of the probability characteristics--the distribution, mathematical expectation, correlation function and spectral density. When random influences are formulated by computer, Gaussian white noise is used as the initial process, passed through linear filters. Statistical interpretation of the output signals is performed using ordinary methods for estimation of the parameters of distributions.

The transfer functions of systems are determined: the error in manufacture of bearing races--the variable drag torque, the error in manufactured--vibration in the axial direction.

USE OF DIGITAL COMPUTER FOR DETERMINATION OF EFFECT OF VIBRATION-DAMPING COATING BY DISPLACEMENT OF RESONANT FREQUENCIES OF STRUCTURE

V. B. Stepanov and V. D. Tartakovskiy (Moscow)

The integral power relationships, relating the displacement of resonant frequencies of a structure when coatings are applied to the change in loss factor are studied experimentally, allowing the loss factor to be estimated on the basis of the calculated or measured frequency shift, the loss factor in turn characterizing the decrease in vibration level.

In this connection, the distribution of vibration amplitudes in various modes of oscillation of the structure before and after application of the coating is studied. The processing of measurement results produced as the dependence of oscillating amplitude on frequency and point of placement of vibrometer was performed by digital computer using an algorithm allowing the statistical mean values of the required parameters, measurement error and calculation error, confidence intervals for the true values of parameters, mean decrease in vibration level and necessary secondary characteristics of the structure to be determined.

FACTORS DETERMINING VIBRATION ACOUSTICAL CHARACTERISTICS OF BALL BEARINGS

B. Ye. Bolotov and V. B. Marnin (Kuybyshev)

Results are presented from a combined study of the influence of the following internal factors on the vibration characteristics of a bearing. Radial clearance, clearance of bearings in separator holes, flat spots and waves on contacting surfaces of internal and external rings and rolling bodies, differences in sizes of rolling bodies. /29

The studies were performed using a commercially manufactured vibration acoustical apparatus and a number of special devices: installations for measurement of vibration accelerations of individual bearing elements, a device analyzing vibration accelerations of the assembled bearing, a semiautomatic device for testing waviness of rolling bodies.

Recommendations are presented for decreasing the level of vibration of general and special purpose ball bearings, and a new vibration acoustical apparatus is described, allowing vibration flaw detection to be performed, indicating the concrete cause of increased vibrations of bearings.

STUDY OF NOISE FORMATION IN THROTTLING DEVICES FOR MEASUREMENT OF NOISE CHARACTERISTICS OF FANS

Ye. Ya. Yudin and N. N. Severina (Moscow)

In this work, we studied throttles of the washer-and-screen type, used in aerodynamic tests of fans. The advantage of this type over other types of chokes (e.g., slide gates, butterfly valves and others) is that they create a distributed resistance which does not twist the flow. They therefore do not create additional turbulence which, particularly in an intake line, may cause additional noise formation in the fan. Furthermore, studies of noise formation in air line elements (A. V. Tolmachev, Ye. Ya. Yudin) have demonstrated that the least noisy elements are those with central placement of the aperture for air transmission. Devices which compress the flow through a slit against one wall of the air line (slide gates, butterfly valves cause additional (7-10 db more) noise generation.

The measurements described were performed using a method and experimental installation described earlier, the diameter of all screens was 150 mm. An apparatus produced by the "Bruel and Kjaer" firm was used. Measurements were performed by the reflected field method with minimum noise level (at night). /30

The range of required change in air stream velocity during the tests was determined on the basis of the range of flow rate factors in general purpose fans (0.2 to 0.8). As applicable to a fan with a blade diameter of 150 mm, the upper limit of velocity was 18.5 m/sec. The minimum velocity in the intake tube was determined by the noise level.

The acoustic power levels of the grids were then compared with the noise spectrum of the fan intake, measured in the same room.

With a flow velocity corresponding to actual conditions (i.e., in this case the velocity in the intake tube of the fan), the total acoustic power level of one grid was 84.5 db. Under these same conditions, the total acoustic power level of the fan (measured in the same room) was 92 db.

A comparison of the noise spectrum of the screen and fan showed that noise formation in the throttling device, with the exception of low frequencies, is significantly lower than the fan noise. Since the fan tested is one of the most quiet (of series produced fans), the difference with other types of fans will be even greater.

METHOD OF STUDYING RELAXATION OSCILLATIONS OF A BALL IN THE SEPARATOR OF A BALL BEARING UNIT

B. Ye. Bolotov and V. V. Trubinikov (Kuybyshev)

The installation diagrammed in Figure 1 consists of the bearing being studied 1 with textolite separator 2, supports 3, on which the inner ring of the bearing is seated and clamp 4 around the outer ring of the bearing. The clamp carries load 5, creating a radial load on the bearing. Brass contact rings 6 and 7 are glued to the separator, and contact carbon brushes 8. The brushes are mounted in bracket 9 made of plexiglass with spring 10. The textolite separator contains brass contacts 11 and 12, connected with rings 7 /31 and 6 respectively. In order to determine the position of a ball in the separator space and its movements with vibrations, the mount around which the bearing is seated is rotated. The electrical circuit consists of a 4.5 V battery, vibrators 13 and 14 of a loop oscilloscope, spring 10, brush 8, rings 6 and 7, contacts 11 and 12 and the ball. The oscillograms produced with various test conditions (load, rotating speed, etc.) can be analyzed to determine the vibration displacements of the ball in the separator. In this /32 manner it was established that at certain moments in time the ball performs relaxation oscillations near the wall of the separator at a frequency of about 200-300 Hz.

Oscillograms of these oscillations are presented and analyzed.

By changing the working conditions of the bearing and its parameters, the vibrations can be reduced or completely eliminated, thus increasing the efficiency of the rolling surface bearings.

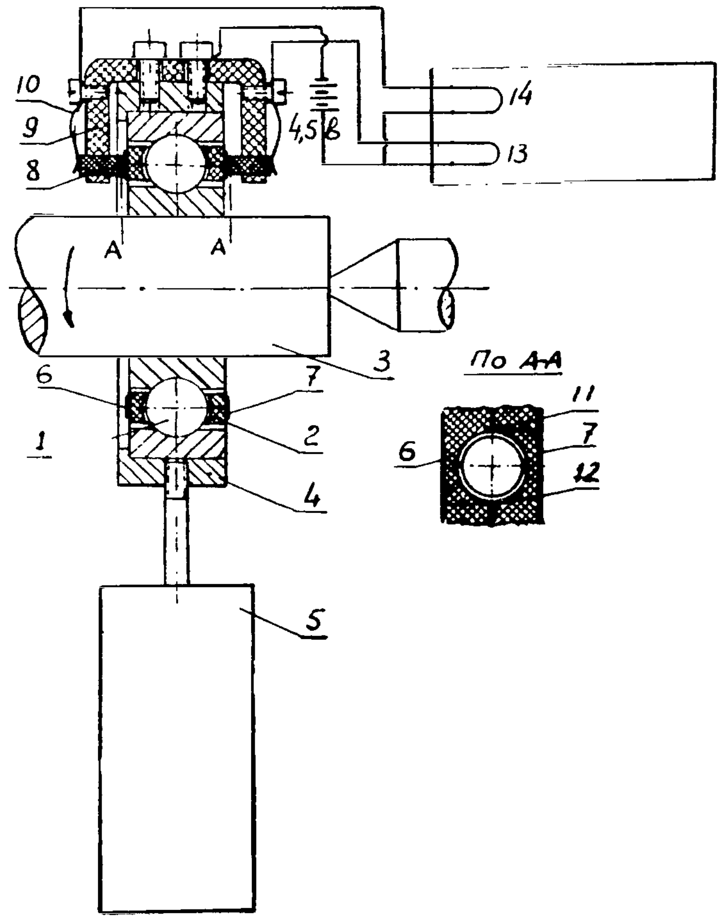

Figure 1.

THE STATISTICAL STRENGTH RESERVE OF DISCRETE SYSTEMS

M. L. Sverdan and Ye. F. Tsar'kov (Chernovitsy)

A method is suggested for calculating the statistical strength reserve of discrete systems, the behavior of which is described by an equation in the form

$$x_{n+m} = \sum_{i=0}^{L} b_j \sum_{k=0}^{m-1} a_{jk} x_{n+k}, \tag{1}$$

where b_j and a_{kj} are constants.

As we know, a necessary and sufficient condition for asymptotic stability of the trivial solution of (1) is:

A) The roots of the polynomial

$$W(z) = z^m - \sum_{j=0}^{L} b_j \sum_{k=0}^{m-1} a_{jk} z^k$$

are located in a circle $|z| < 1$.

This condition is considered fulfilled throughout the following.

Theorem 1. The statistical strength reserve S_j with respect to parameter b_j is

$$S_j = \left[\frac{1}{2\pi i} \int_{|z|=1} \frac{\left(\sum_{k=0}^{m-1} a_{kj} z^{-k}\right)\left(\sum_{k=0}^{m-1} a_{kj} z^k\right)}{zW(z)\, W(z^{-1})} dz \right]^{-1} \tag{2}$$

Theorem 2. If the coefficients b_j of equation (1) are excited by the independent sequences α_{nj} with zero means and variance σ_j, the trivial solution of the excited system is asymptotically stable in the mean square if and only if /33

$$\sum_{j=1}^{L} S_j^{-1} \sigma_j < 1. \tag{3}$$

Theorem 3. If the random quantities α_{nj} indicated in theorem 2 have variances σ_{nj}, where $\lim_{n\to\infty} \sigma_{nj} = \sigma_j$, then (3) is a sufficient condition for mean square asymptotic stability of the trivial solution of the excited system.

This report presents conditions for various stable modes of simple vibration-impact systems on the assumption that the coefficient of restitution upon impact is a random quantity.

INFLUENCE OF METHODS OF FASTENING OF ACCELEROMETER ON ITS FREQUENCY CHARACTERISTICS

Yu. M. Vasil'yev, Yu. K. Konenkov and L. F. Lagunov (Moscow)

Based on an analysis of an accelerometer considering the rigidity of its mounting, an analytic expression is produced for the necessary set frequency of the converter (ω_{set}) depending on its natural frequency (ω_0), required to provide a frequency characteristic with fixed unevenness over a predetermined frequency range

$$\omega_{set} = \frac{\omega_b^2 - \omega_n^2(1+\varepsilon)}{1 - \frac{\omega_n^2 \Delta}{(1+\Delta)\,\omega_b^2}}$$

where ω_b is the upper boundary frequency of the range used, up to which the unevenness of the frequency characteristic does not exceed Δ,

ε is the ratio of the mass of the inertial element to the mass of the body of the accelerometer.

The expression produced can be used to determine the necessary value of elasticity of mounting of a converter to an object for measurement.

The frequency characteristics of accelerometers with various mounting methods are studied experimentally in this work.

The influence which intermediate fastening elements such as collars, bars and other devices used when the sensor cannot be installed directly at the required point on the object have on the frequency characteristic of sensors is studied analytically. /34

An experimental device is recommended for investigation of the influence of intermediate sensor fastening elements.

STUDY OF COLLISIONS OF MACHINE ELEMENTS ON THE BASIS OF PHENOMENOLOGICAL MODELS OF INELASTIC MEDIA

A. N. Lenskiy, V. M. Loboda and L. P. Fabrika (Dnepropetrovsk)

The experience gained in application of analog models for determination of the force modes of operation of elements of heavy machines in the mining and metallurgical industry is discussed. The general problems involved in the statement and solution of the problem of determining impact parameters (duration, magnitude and form of impact pulse, impact energy, coefficient of restitution), as well as problems of determining the dynamic results of impact loads between the elastic links of machines resulting from impact interaction of masses are presented. A method is described for determining areas of stability of periodic modes of motion of impact and vibration-impact systems using analog models.

A generalized model for substitution of an elementary volume of a medium is suggested. The colliding bodies are considered as consisting of a set of elementary volumes. Each elementary volume and its coupling to neighboring volumes are replaced by a rheological model, the properties of which are determined by the properties of the material of the colliding bodies. Rheological models of viscoelastic, elasto-plastic and viscoelastic-plastic media are studied. It is demonstrated that these models describe the properties of structural materials used in machine building sufficiently completely (for purposes of modeling of collisions).

The characteristics of the substitution models are calculated or determined on the basis of results of static and dynamic tests of materials.

The mathematical models suggested describe the collisions of bodies, the period of natural oscillations of which is short in comparison to the duration of the collisions. The use of the models suggested for the study of collisions of machine elements is justified.

Examples are presented of modeling concrete impact and vibration-impact systems: a vertical vibration transporter, an unbalanced rotor, rotating in a bearing without lubrication, and a Geneva mechanism. The results of modeling are compared with precise analytic solutions. /35

METHODS OF STUDYING VIBRATIONS OF HIGH-SPEED DIESELS

L. V. Giuzov, M. A. Miselev and I. M. Chirkov (Leningrad)

Determination of the vibration levels of high-speed diesels by calculation required determination of complex mechanical resistances (impedances) of individual parts of the diesel, its combined units, shock absorbers and foundation. At the present time, mechanical resistance is calculated only for rods with definite boundary conditions. For parts with more complex configuration with variable cross sections such as crankshafts, blocks, crankcases, etc., mechanical resistances are not calculated. The difficulty of the problem is that when a single type of oscillation is excited, e.g., longitudinal oscillations, further propagation is accompanied by the development of flexural oscillations and vice versa. Furthermore, in the high frequency area there are many possible resonant phenomena, which are extremely difficult to consider in advance.

Therefore, experimental studies are first required, generalization of the results of which would make possible the application of a definite mathematical apparatus. Studies were performed with a standard high-speed, lightweight V-12 diesel type ChN 18/20 with an effective power rating of 1200 hp at 1850 rpm with a level of vibration of support lugs of 118 db (acceleration) and an air noise level of 125 db. Measurement of combined mechanical resistances was performed in the mode of stable sinusoidal oscillations for the following diesel parts and units: block, upper and lower crankcases, pistons, crankshafts, connecting rods and assemblies. The point and transfer impedances were determined in the 20 Hz-10 KHz frequency range.

The measuring apparatus included a channel for measurement of oscillating speed with a filter and phase inverter and a channel for measurement of oscillating power with a filter. The voltage from the outputs of each channel was fed to a phase meter or to the plate of a cathode oscillograph for measurement of the phase shift angle between force and speed. An impedance head was /36
made according to a design developed by N. N. Kupriyanov and I. L. Orem, with several design improvements, allowing the accuracy of measurements in the high frequency area to be improved. The source of excitation used was an electrodynamic vibrator for the area of low and middle frequencies and a piezoelectric

vibrator for the high frequency area. Calculation of the impedance modulus, its imaginary and real parts was performed using known formulas.

DIFFERENTIAL EQUATION FOR HORIZONTAL LOOP OF SIGNAL CARRIER

P. A. Varanauskas (Kaunas)

In magnetic recorders using free tape loops, an impact occurs as the tape is fed. This causes noise and vibration of the tape, which has a negative influence on the accuracy of magnetic recording and reproduction of signals in tape drive mechanisms.

It has been noted that the impact is decreased if the walls are curved. The optimal radius of curvature corresponds to the curvature of the tape loop. defined by the differential equation

$$M = EI \frac{\frac{d^2y}{dx^2}}{\left[1+\left(\frac{dy}{dx}\right)^2\right]^{\frac{3}{2}}}$$

and the equation

$$M = a_1 y$$

where a_1 is the force of compression of the ends of the loop.

Solving, we obtain

$$X = \pm\left[\frac{2}{a}\sqrt{\frac{4-c^2}{8+2c^2}} - \frac{1}{a}\sqrt{\frac{(8+2c^2)(2-c)}{2+c}}\right]\int_0^u \frac{d\varphi}{\sqrt{1-\frac{2-c}{8+2c^2}\sin\varphi}} \pm$$

$$\pm \frac{1}{a}\sqrt{\frac{(8+2c^2)(2-c)}{2+c}}\int_0^u \sqrt{1-\frac{(2+c)^2}{8+2c^2}\sin^2\varphi}\, d\varphi .$$

Here /37

$$U = \sqrt{\frac{2+c}{a}(1-\sin^2\varphi)} = \sqrt{\frac{2+c}{a}}\cos\varphi .$$

The equation has the form of elliptical integrals. This is easily solved by tabular methods or by computer.

STUDY OF FLEXURAL OSCILLATIONS OF MAGNETIC DRUMS

Yu. Yu. Getsevichyus, Z. I. Potsyus and K. M. Ragul'skis (Kaunas)

Problems of the flexural oscillations of magnetic drum recording media are studied, as well as means for reducing and measuring them.

The primary sources and causes of oscillations in the angular velocity are determined for various types of magnetic drums. The evenness of the velocity of electric drive motors used to drive magnetic drums is studied in detail. Investigations are performed using various methods for stabilizing the rotation rate. Efficient designs for drive mechanisms with increased rotation smoothness are suggested.

The oscillations in angular velocity were measured using tandem piezo-sensors of universal vibration measuring apparatus, or by a frequency method involving comparison of the phase of periods of standard and measured pulse sequences.

The data from experimental studies were processed by a statistical method on a computer.

VIBRATIONS OF A STORAGE DRUM OF A BESM-6 COMPUTER

V.-R. V. Atstupenas, Yu. Yu. Getsevichyus, I-M. P. Pakaushite and K. M. Ragul'-skis (Kaunas)

Vibrations of a memory drum influence the constancy of the recording spacing and the rate of movement of the recording surface of the drum in relation to the magnetic heads. This variable component in relative velocity is added /38 to the circular velocity of the drum. A change in the spacing during operation of a memory device causes parasitic amplitude modulation of the signals recorded and read. Changes in relative velocity, as well as vibrations of the magnetic drum in the axial direction, result in parasitic phase modulation of the signal.

An experimental study of drum vibration was performed using a high-sensitivity multichannel measurement apparatus and contactless capacitive sensors developed for the purpose. The results of recording of drum vibrations were processed by statistical methods on a computer. The principal sources of vibration of a magnetic drum are determined. Measures to decrease the vibrations of the magnetic drum are indicated and studied analytically.

STUDY OF VIBRATION AND DYNAMIC BALANCING OF MECHANISMS, INCLUDING SEVERAL PARALLEL SHAFTS

I. Yu. Yurgaytis, Rem. A. Ionushas and K. M. Ragul'skis (Kaunas)

The possibility is studied of balancing rotating objects in an assembly with several parallel shafts in the form of a gear box used, in particular, in precision machine tool building.

Differential equations of motion are derived for a system including several parallel shafts rotating at various angular velocities.

Solution of the equations of motion of the system is used as a basis for deriving analytic dependences between the parameters of unbalanced rotating shafts and the oscillating parameters of the system. This allows determination of the magnitude and point of imbalance for all shafts simultaneously.

Results are presented from experimental studies confirming the theoretical assumptions. Analysis is performed of vibrations of gear box mechanisms. Oscillograms of the vibrations are processed statistically by computer. This reveals that the graphs of spectral power are dominated by frequencies excited by the unbalanced rotating shafts.

ANALOG COMPUTER STUDIES OF OSCILLATIONS OF SYSTEMS HAVING HYSTERESIS CHARACTERISTICS

Yu. A. Kozlov, V. F. Rakhmanov and A. A. Samarin (Moscow)

The methods and results are presented from a computer study of dynamic processes in systems having hysteresis characteristics. The authors have developed an optical-electronic functional converter (OFC) consisting of an attachment to an ordinary analog computer for reproduction of hysteresis characteristics during modeling. The operation of the OFC is based on the use of the ambiguous dependence of photocurrent of several types of industrial photoresistors on applied voltage, which has the form of a hysteresis loop with branches contacting at the coordinate origin, with a natural counterclockwise course. /39

Changing the level of light flux and its distribution over the sensing surface of the photographic emulsion are used to regulate parameters of the loop during selection of the required characteristics of the oscillating system being modeled. The form, area and degree of asymmetry of the hysteresis loop are regulated. Connection of the OFC in the input circuit and the feedback circuit of the operational amplifier allows the direction of movement of the loop contour to be changed. This allows modeling of dynamic processes considering both accumulation and dissipation of energy in oscillating systems, resulting from the presence of hysteresis properties. Some of the characteristics reproduced using the OFC are shown on Figure 1 (U_1 is voltage at input of OFC, U_2 is voltage at output of OFC).

When using the OFC, the analog computer can solve nonlinear differential equations, one or more terms of which are ambiguous hysteresis dependences of this type:

$$a_n x^{(n)} + a_{n-1} x^{(n-1)} + \ldots + a_1 \dot{x} + a_0 f(x, \dot{x}) = F(t). \tag{1}$$

The symbols used here are: a_n are constant coefficients, $f(x,\dot{x})$ is the hysteresis characteristic, $F(t)$ is the external excitation.

The solution of an equation such as (1) was studied where $n = 2$, $a_1 = 0$ and $F(t) = F(t + \tau)$, where τ is the period of oscillations produced by analysis of forced oscillations of mechanical systems considering damping by internal friction, as well as self-oscillations caused by internal friction. It is demonstrated that the use of the method suggested for reproduction of hysteresis characteristics for analog computer modeling of such problems provides new capabilities for increasing the reliability and durability of the significant parts and units of machines and apparatus subjected to vibration.

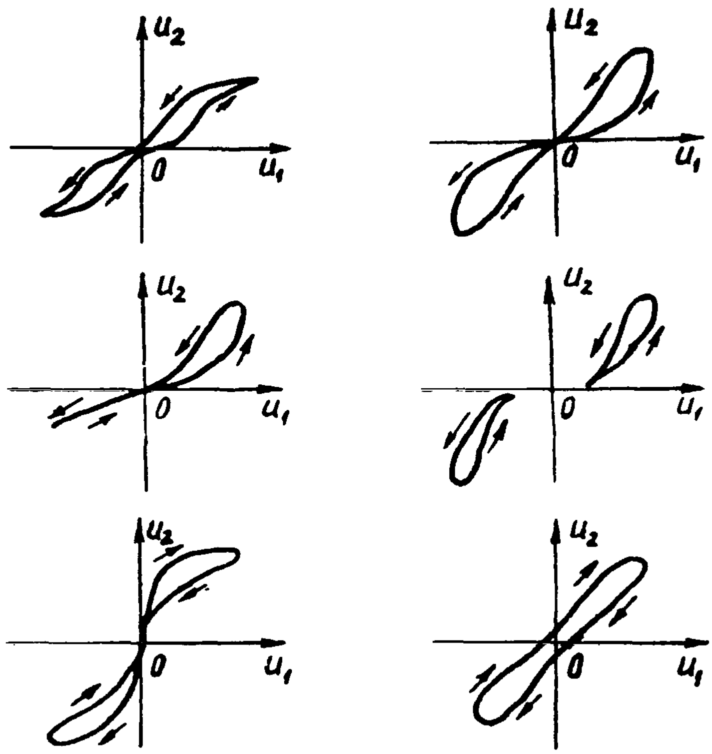

Figure 1.

ELECTROMAGNETIC IMPACT MACHINES

N. P. Ryashentsev (Novosibirsk)

Theoretical and experimental studies were performed on the following problems in order to aid in the creation of electromagnetic machines with increased power characteristics:

-- transformation of electric energy to magnetic energy, then magnetic energy to mechanical work; /41

-- power supply, control and protection system;

-- calculation of magnetic system;

-- electromagnetic transient processes;

-- determination of optimal parameters of machines;

-- development of principles of the construction of automatic control systems for operating processes;

-- thermal processes, reliability and durability of machines.

At the present time, the electromagnetic system has been little studied for reciprocating engines. The electromagnetic transient processes occurring in the system are described by nonlinear differential equations, during the composition of which it is difficult to consider saturation of a magnetic circuit, mechanical losses, the influence of eddy currents, hysteresis losses and a number of other factors which significantly influence the operating mode of a machine. The effect which eddy currents and block drag have on the operating process of a machine is difficult to consider due to the small amount of study to which these processes have been subjected.

Theoretical and experimental studies performed have allowed the influence of certain system parameters on the working process and the power characteristics of an engine to be established. The results of investigation were used in developing impact units for impact engines of 0.2, 0.4, 1.0 and 60 kg with an impact frequency of 1500 and 3000 per min, designed for impact and impact-rotary machines. The efficiency of the impact units developed reaches 43-45%. An impact unit has been tested with an impact energy of 200 kgm and a frequency of 50-60 impacts per min, designed for crushing of odd sizes; a range of impact units operating at various energies and impact frequencies and reciprocating engines are being developed.

ESTIMATION AND NORMALIZATION OF OPERATING MODES OF VIBRATION TECHNOLOGICAL MACHINES

K. A. Olekhonovich (Poltava)

Estimation of the dynamic mode of vibration technological machines (vibrating platforms, vibration mixers, vibrating mills, vibrating tumbling machines, etc.) is usually based on the amplitude of movement and the oscillating frequency of the working element.

With this approach, it is difficult to establish a direct dependence between the intensity of vibration and its technological productivity for /42
machines of similar purpose, but using different type of oscillations (circular and directed harmonic, vibration-impact, vibration-shaking, etc.)

Since a fixed technological effect of vibration processing, with otherwise equivalent conditions, is provided basically due to the influence of forces of inertia (their magnitude, direction, nature of application), and the actuating element of a vibration machine can be looked upon as a generator of these forces, the intensity of the vibration mode can be evaluated by the work of forces of inertia per second, related to each unit of oscillating mass.

This criterion is called the relative power of dynamic effect and can be established analytically for the types of vibration machine listed as a function of their design parameters and operating modes.

Direct measurement of the criterion during the operation of a vibration machine seems possible using the proper apparatus.

Since this criterion reflects the intensity of energy of the vibration processing mode, the technological effectiveness of a vibration machine and its durability are proportional to this criterion and the operating time, which should allow automation of production of processes involving vibration, and also should allow determination of optimal maintenance intervals for vibration machines.

ANALYSIS OF THE DYNAMICS OF MULTI-MASS ELASTIC SYSTEMS BY A FREQUENCY METHOD

S. S. Ivanov and V. F. Preys (Tula)

A grapho-analytic method is described for calculation of chain and branched multi-mass elastic systems (MES) with regard to damping under steady-state conditions, based on the use of logarithmic frequency responses.

MES are described by a system of related linear second order differential equations for which the right-hand side consists of semi-logarithmic functions of time. Conversion of this system of differential equations allows a transition to the construction of structural plans for MES in the form of transfer functions of typical links and the construction of logarithmic MES frequency characteristics.

Cumbersome calculations are replaced by graphic constructions performed /43
using templates and V. V. Solodovnikov monograms. In this manner, MES with more than four masses can be calculated with several semi-logarithmic perturbing actions on the system, in cases where analytic calculations are quite cumbersome. The task of synthesis of a branched MES of a machine is significantly simplified.

STUDY OF REPEATED IMPACT INFLUENCE OF PRESSURE ROLLER ON MAGNETIC TAPE

Ye. V. Rozanov and Yu.-K. V. Rozenkrants (Moscow)

In start-stop tape drive mechanisms, the repeated impact effect of the pressure roller on the magnetic tape causes tape breakage, decreasing the reliability of reproduction of information. Theoretical studies have shown that the breakage of magnetic tape depends on the friction force arising between the elements in the drive system and the tape during a start. Experimental studies have demonstrated the relationship between the decrease in level of signal reproduced and tape drive mechanisms causing friction. Recommendations are presented for the development of tape drive mechanisms with repeated impact effect of the pressure roller against the magnetic tape.

DIAGNOSIS OF STATES OF PLANETARY GEAR REDUCER FOR CERTAIN PARAMETERS

F. Ya. Balitskiy and A. G. Sokolova (Moscow)

The problem of acoustical diagnosis of gear drive systems is studied as applicable to an experimental planetary reducing gear designed by the authors. As parameters of the state of the gear system, the play and loads upon engagement were studied. The statistical characteristics of the vibration processes (in discrete form--multivariate vectors) or the various parameters of state served as patterns for recognition. The cosine of the angle between the vectors determines the distance between patterns.

Tables of coincidents of the parameters of the states and these quantities are produced, allowing diagnosis of states for the parameters studied.

POSSIBILITY OF TESTING THE TECHNICAL CONDITION OF INTERNAL COMBUSTION ENGINES ON THE BASIS OF NOISE AND VIBRATION PARAMETERS

V. N. Lukanin (Moscow)

The equations describing the forces in a crankshaft-connecting rod mechanism contain no indications on the impacts which can occur, for example, in bearing A. The fact of an impact and the subsequent excitation of parts can be established by using the concept of the dynamic reaction and by introducing a so-called massless link to the crankshaft-connecting rod mechanism, replacing the clearance in the bearing and allowing its influence on the forces of the interaction between bearing elements to be considered. Analysis of the expressions for dynamic reaction indicate that it vanishes at certain crankshaft rotation angles, which indicates a loss of power closure, a subsequent impact in the bearing and development of vibration and noise. The parameters of the vibrating pulse, in particular the maximum magnitude of oscillations, depend on the clearance in the joint, and can be used for diagnostic purposes. /44

POSSIBILITY OF ESTIMATING WEAR BY VIBRATION-ACOUSTICAL METHODS

A. A. Skuridin and G. S. Sabadash (Leningrad)

Theoretical and experimental investigations are performed into the dependence of vibration of a D6 diesel and its mechanical noise on the clearance between the piston and cylinder liner of this diesel. It is demonstrated that the vibration of the cast iron block of a D6 diesel in the 2955-3045 Hz range (f_{av} = 3 KHz) and in the 6895-7105 Hz range (f_{av} = 7 KHz) at nominal speed and nominal fuel feed per cycle, and also with two cylinders next to the cylinders being measured disconnected, increases in proportion to the increase in hot clearance resulting from wear of the liner face and piston skirt (see figure). Having a graph of this relation $L_{ecc} = f(\delta)$ and measuring the /45 vibration of the block of another D6 diesel, it is possible to determine the clearance between piston and cylinder lining, i.e., to judge the wear of this friction sliding pan.

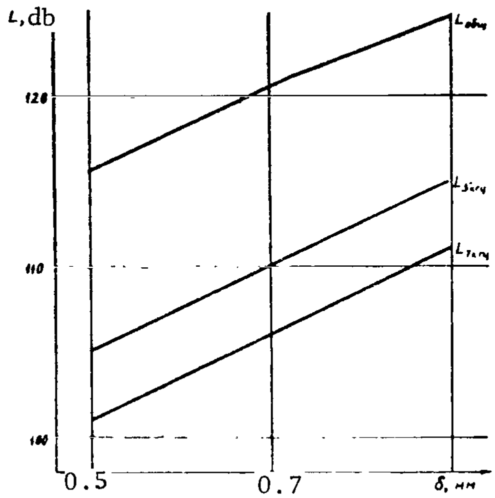

Figure 1. Change in Vibration of Cast Iron Block of D6 Diesel Near Cylinder 6 as a Function of Change in Clearance of Piston at Nominal Rotating Speed with Nominal Fuel Feed Per Cycle (Cylinders No. 4 and 5 Operating). Width of frequency band = 3%.

DIAGNOSIS OF REAR AXLE REDUCTION GEAR OF ZIL-130 MOTOR VEHICLE BY ACOUSTICAL METHOD

M. P. Kocehv (Moscow)

With the unit on a test stand, the acoustical characteristics of channels and operating modes were determined and the influence of oil viscosity on oscillation parameters was estimated.

For the primary couples of the unit, which characterize its reliability, quantitative estimates are produced of the relationships between structural parameters and the acoustical signal.

An estimate of the technical condition of the reduction gear directly on the vehicle, operating on a test stand with drums beneath the wheels, indicated that the results of test stand experiments will require correction.

The technological condition of a reduction gear under actual operating /46 was determined by information contained in the total level of oscillations with parallel recording of oscillations on magnetic tape and by structural parameters produced by micrometry of parts of the unit.

The range of change of acoustical signals characterizing the technical condition of the primary couples in the reduction gear is 9-14 db.

A comparison of the values of an acoustical signal with structural parameters allowed test indices for diagnosis of the reduction gear to be determined.

USE OF STATISTICAL METHODS IN THE INVESTIGATION OF VIBRATIONS AND THE DYNAMICS OF MECHANISMS AND MACHINES

S. P. Kitra and K. M. Ragul'skis (Kaunas)

Statistical methods were used to study the process of oscillations in the tension of a magnetic tape as one of the primary parameters of a magnetic tape drive, determining both the frequency-amplitude modulation and other specific distortions of the signals reproduced.

Analysis of the estimates of the correlation functions produced (Figure 1) of the process of unevenness of tension produced on the basis of realizations recorded at different times gives us reason to assume that the process is not an ergodic stable process in the strict definition.

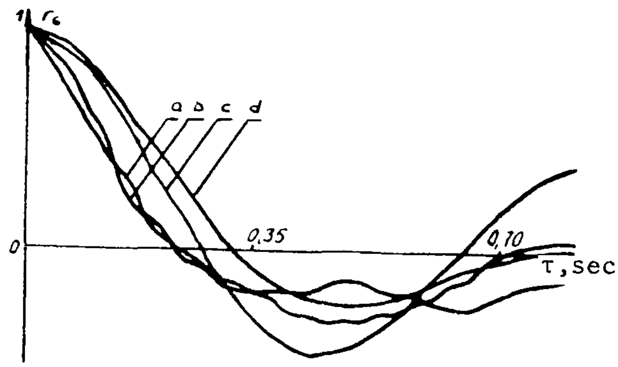

Figure 1.

The mutual correlation functions at characteristic points on the line /47 characterize the interrelationship of processes occurring. The shift of the maximum ordinate of the mutual correlation function (Figure 2) to the right by t_m along the time axis characterizes the delay and time constant of the object. The sector of estimation of mutual correlation functions falling in the area of negative τ is distorted due to the presence of explicit or implicit feedback in the object. The property of a random signal that it changes when passing through a linear system is a basis for identification of the dynamic characteristics of units of the mechanism. However, determination of the dynamic characteristics of the object by this method is hindered by a number of factors: a certain degree of correlation of the effective perturbations, the presence of explicit and implicit feedback, the absence of full stability, the existence of certain nonlinearities of statistical and dynamic characteristics, etc.

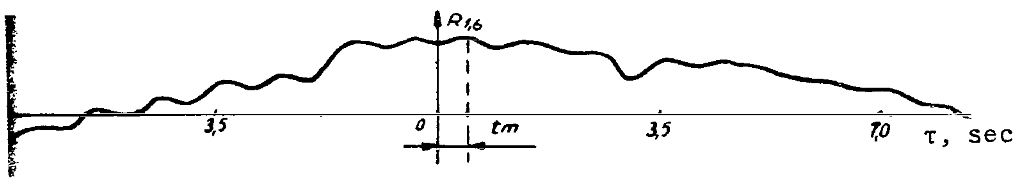

Figure 2.

OPTIMIZATION OF THE CROSS SECTION OF THE FRAMES OF A MULTISTAGE VIBRATING STAND

A. P. Sibrin (Chelyabinsk)

Using the example of a three-stage vibrating stand--a gimbal support (Figure 1), a method is described of determining the optimal relationship of the dimensions of the cross section of the frames of a multistage vibrating stand, allowing the dimensions of the cross section to be determined with fixed minimum natural oscillating frequency of the frames ω at the points of application of the useful load or the next frame stage, with provision of maximum rigidity C and minimum moment of inertia of the frame with useful load I installed on it.

Corresponding expressions are presented for concrete types of frame cross section; in particular, it is demonstrated that the optimal value of β, equal to the ratio of the width of the cross section B to the height of the frame H, and the value of B are unambiguously determined by the natural oscillating frequency of the frame. /48

These curves can be constructed for each frame of a stand with various cross sectional forms.

The method suggested also allows the optimal form of frame cross sections to be determined.

A SELF TUNING CONTROL SYSTEM FOR THE MOTION OF A VIBRATING STAND

A. A. Koshcheyev and A. P. Sibrin (Chelyabinsk)

In physical modeling of complex dynamic systems, the problem arises of creating multistage vibration stands, reproducing with the required accuracy the perturbing actions actually acting on the system being studied.

Using a two-stage vibrating stand with an electrodynamic vibrator as an example, the principles involved in the creation of a control system of its motion are analyzed.

In order to provide the required accuracy of reproduction of input actions, control must be performed in a closed cycle. As a similarity criterion and, consequently, method of provision of feedback, the similarity of the spectral densities of the desired and actual stand movements is selected.

The latter can be performed using a self-tuning system with extremal adjustment of the correcting circuits (Figure 1).

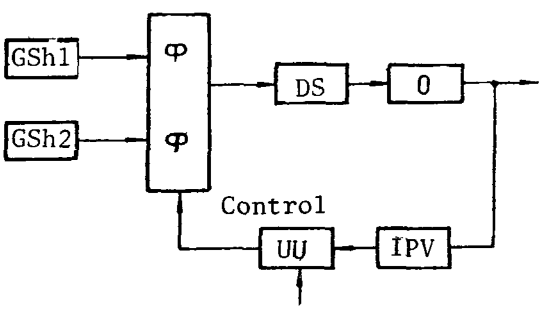

Figure 1.

Analysis of the motion control system of the vibrating stand produces the equation /49

$$\bar{d} = A\bar{k}, \qquad (1)$$

where $\bar{d}$ is the assigned dispersion vector;

$\bar{k}$ is the vector of controlled coefficients;

A is the quadratic matrix of coefficients, depending on the frequency characteristics of the filters and the control object in the corresponding frequency band.

This equation, with random values of the elements of matrix A, can be solved using an iterative method of successive approximations. The algorithm for solution of equation (1) thus becomes

$$\bar{K}_i = \bar{K}_{i-1} + \frac{2}{\mu}\bar{e}_i, \qquad (2)$$

where $\overline{K}_i$, $\overline{K}_{i-1}$ represent the i-th and i - 1th approximations of the vector of the controlled coefficients, respectively.

$\overline{e}_i$ is the error at the ith step;

μ is the first norm of matrix A.

A possible plan for a hardware realization of the algorithm (2) for one channel is presented (Figure 2).

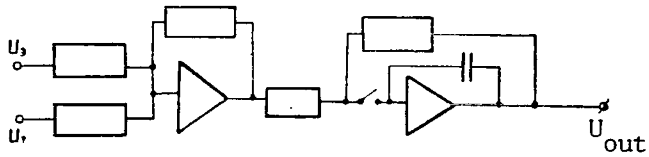

Figure 2.

COMPENSATION OF THE INFLUENCE OF AN ELECTRODYNAMIC VIBRATING STAND ON CHARACTERISTICS OF MECHANICAL STRUCTURES PRODUCED IN VIBRATION TESTS

L. A. Manashkin and A. M. Tikhomirov (Dnepropetrovsk)

A method is presented for constructing a compensator for the mutual influence of an electrodynamic vibrating stand and a test object, and the results are presented from studies of certain characteristics of the compensator-stand system. The algorithm for construction of the compensator is independent of the test object and is determined entirely by the mathematical description of the stand and measurement systems selected. The compensator is an automatic control system with feedback. The compensator developed consists of two parts: the /50 compensator for the electric portion of the test stand with current feedback in the moving coil of the vibrator and the vibrator compensator with feedback based on the acceleration of the vibrating table.

The compensator for the electric portion of the stand significantly decreases the influence of the test object on the dynamic characteristics of the electric portion of the stand. In particular, if the force developed by the vibrator is taken as the output quantity, it can be considered that it is independent of the oscillating frequency and the test object.

The compensator of the vibrator significantly increases the influence of the test stand on the dynamic characteristics of the test object.

The compensator was produced using the operational amplifiers of an EMU-8 analog computer.

The investigation of the operation of the compensator was performed with the vibrating stand loaded by objects with linear and nonlinear dynamic characteristics, with various input actions (monoharmonic, polyharmonic, random stable, unstable) and with various noise levels in the feedback channels of the compensators.

LOW FREQUENCY VIBRATING STAND FOR PHYSICAL MODELING OF THE MOVEMENTS OF COMPLEX DYNAMIC SYSTEMS

G. S. Chernorutskiy and A. P. Sibrin (Chelyabinsk)

The principles of construction of low frequency vibrating stands for physical modeling of the motions of sensing elements in control systems for moving objects are studied, including stands for investigation of the functions of the vestibular apparatus, which, as we know, is the biological regulator of the position, motion and orientation of man in space.

Examples are presented of the structure of vibration stand motion control systems, specifics of their operations and certain kinematic diagrams.

It is demonstrated that analysis of the control systems of the motions of vibrating stands requires that the random nature of the parameters be considered.

Methods are presented, allowing synthesis of systems controlling the motions of a stand to be performed considering the random nature of the parameters.

OPTIMALIZING INPUT CONTROLLER OF OSCILLATING AMPLITUDE FOR FATIGUE TESTING OF PARTS

V. I. Yermolin and N. Ye. Salamatin (Kazin')

An automatic control system regulating the oscillating amplitudes of parts during fatigue tests on electrodynamic vibrating stands with simultaneous tracking of changes in the natural frequency of the part being tested and the corresponding adjustment of the frequency of the exciter unit has been developed. /51

This system allows the following principal parameters to be recorded during the testing process:

a) Recording of changes in amplitude and frequency on strip chart;

b) Recording of number of oscillating cycles of part being tested.

The electrical circuit of the device differs in principle from existing similar devices. The system uses electronic regulation in place of electromechanical.

The system has the following technical characteristics:

1. Material of parts tested--nonmagnetic.
2. Operating frequency range--55-3000 Hz.
3. Absolute mechanical stresses correspond to output voltage of 0.15-150 mv.
4. During the tests, deviations from a fixed amplitude level are recorded by a strip chart recorded with a reading accuracy of at least ±0.5% of the stabilized level.
5. The accuracy of maintenance of the fixed voltage level is at least ±2%. The time required to reach the test load is not over 2 sec.
6. The type of sensor used is a wire sensor (strain-gage sensor) with a resistance of 1000 ohms and an induction sensor.

In case of a failure of a strain-gage sensor, the system automatically goes over to operation by induction without stopping.

7. The permissible imbalance of phases of oscillations in the resonating portion of the part and vibrator table during operation of the installation is not over ±3%. The error in recording a deviation in frequency is not over ±0.5 times the natural initial frequency of the part.

8. The capacity of the cycle counter is eight digits with visual readout of the last five digits.

9. The system disconnects the vibrating stand amplifier when an output power of 0.9 P_{max} is reached, and also in case of a change in exciter frequency of any specimen of 5-20% of the initial established frequency.

ANALYSIS OF MECHANICAL SYSTEM WITH MATCHED VIBRATOR

N. A. Karpov and V. N. Lobanov (Moscow)

Four plans for vibration-impact systems with two masses with one or two elastic forces are studied with matched self-synchronizing imbalance vibrator, which can be placed on the upper or lower mass. With oppositely directed rotation of the imbalanced stator and rotor of the vibrator, a co-phased synchronization mode is possible. The stability of the steady state is studied, assuming for the angular coordinates $\phi_1 = \tau + \psi_1, \phi^{\bullet}_2 = \tau + \psi_2$, where $\psi_{1,2}$ are slowly changing functions of time. Four systems are compared as to degree of stability of the co-phased synchronization mode (with decreasing modulus of roots of the characteristic equation, the degree of stability increases). The case is analyzed when the matched vibrator is isolated from impacts. /52

MULTICHANNEL APPARATUS FOR DIAGNOSIS OF VIBRATING MODE OF MAIN SHIP ENGINES

N. Prokhorov (Taganrog)

For proper selection of the operating mode, an engine operator must process a large volume of information. However, even an experienced operator, in evaluating the level and nature of vibrations of engine parts, may not succeed in making the proper diagnosis or may make an error. In both cases, this can lead to an emergency, injury, etc. Under these conditions, it is necessary to increase the reliability of conclusions concerning the vibrating mode of operation of an engine. This is being done by transmitting an ever greater share of the diagnostic functions of the operator to instruments.

This report presents the principles, circuits and technical data used in testing and signaling apparatus developed at the "Vibropribor" Special Design Bureau. This apparatus allows:

a) Indication of the current value of vibration acceleration;

b) Light signaling when a fixed level of vibration acceleration is reached at any one of 24 points;

c) Output of a signal to an external signaling device indicating which of the 24 points has seen a change in vibration level.

Information on vibration of engine parts produced from the apparatus allows the operator to evaluate the operating mode of the engine and make a proper decision. /53

In the new apparatus suggested, the information produced is converted to a form convenient for processing by various recording and analyzing instruments, and also for processing by computer.

The apparatus developed, together with the instruments, informing the operator concerning other engine operating parameters, can be included in any combination of automatic or semiautomatic engine control devices or devices for controlling any other machine.

PROBLEMS OF THE DYNAMICS AND STABILITY OF A CYLINDRICAL SOLID ON AN AEROSTATIC SUPPORT

L. A. Bushma (Kaunas)

The suspension of a cylindrical body on an aerostatic support is analyzed. For this purpose, a diagram of a power supply is developed, and three test stands are constructed and evaluated. The first stand has the highest frequency of free vertical oscillations (over 10 Hz), the second has a lower frequency of vertical oscillations, and the third has a variable resonator to suppress weak self-oscillations as they arise.

The work contains the results of a study of the dependence of the height of floating on load at various pressures, determines the rigidity of an aerostatic support, its dependence on load and pressure. It is also determined that aerostatic elastic couplings are nonlinear in nature and their elasticity has a rigid restorative force. Free and forced oscillations and the influence of individual parameters on the operation of the system are determined. The system is studied with asymmetrical load. The theoretical study of the behavior of this system is performed using closed and open aerostatic support. The differential equations of motion are composed, and analytic dependence is presented between pressure and the volume of the lubricating layer. Oscillo- /54 grams of the various motions of the system are presented. Investigations using analog computers were performed.

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STUDY OF VIBRATIONS OF A MAGNETIC DRUM WITH PNEUMATIC DRIVE AND PNEUMATIC SUSPENSION

Yu. Yu. Getsevichyus, K. M. Ragul'skis and B. -Yu. B. Yanchyukas (Kaunas)

A model of a magnetic drum having high vibration resistance and compactness as well as very low level of oscillations is studied. The drum itself consists of a thin ring mounted on a nonmoving pin with a clearance of a few tenths of a micron, rotating on radial air and journal bearings with external forced air and successive feed. The air streams of the bearing itself are used as the drive to apply torque to the drum.

There has been developed a method for the precise measurement of oscillations of the magnetic drum by contactless sensors, as well as a method for measurement of the static rigidity, pulsations of air in the hollow pin, torque acting on the magnetic drum and angular oscillations of the drum.

The experimental data are processed by mathematical statistics using digital computers. The spectral densities and correlation functions of drum oscillations are produced as a function of external blowing pressure, mass and diameter, location and angle of inclination of the air supply apertures. The influence of the macro characteristics of the external surface of the pin, and also of the internal and external surfaces of the drum itself and its oscillations, is determined.

Oscillations of a magnetic drum with pneumatic drive and pneumatic suspension and oscillation of magnetic drums of similar size but rotating on precision ball bearings by an electric motor are comparatively analyzed.

VIBRATION DAMPING OF BEARING UNITS

A. F. Koz'yakov and Ye. Ya. Yudin (Moscow)

In order to reduce the oscillations, it is suggested that bushings be installed between the body of a woodworking lathe and the external ring of the bearings of the cutting tool (in particular a cutting shaft). /55

In order to determine the optimal form and material of the bushings, a number of special experiments were performed. First of all, the task was set of determining the mechanism of attenuation of oscillations during placement of the bushings. Bushings 10 mm thick were tested, made of bronze, aluminum, textolite and capron. The first two materials have high values of the reactive portion of the impedance, i.e., act like ordinary inserts. Capron and textolite have high internal friction factors and consequently significantly active impedances, so that they absorb oscillations intensively. The experiments performed showed that the use of bronze in aluminum bushings does not result in any reduction in vibration level and related noise of spindle units of woodworking machines. However, textolite, and particularly capron bushings result in a significant reduction in vibration level of the system made up of the tool and workpiece, which is greatly important if we recall that most woodworking machines are hand fed.

Measurements were performed using a rough planer type SF4-4 operating at n = 3000 rpm both in operation and when idling.

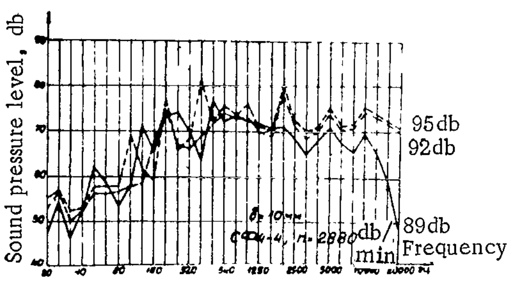

Figure 1. Noise Spectra of Spindle Units With Vibration-Damping Bushings. 1, No bushing; 2, Textolite bushing; 3, Capron bushing.[1]

[1] Russian text does not indicate the number of the lines.

Figure 1 shows the noise spectra of idling of the SF4-4 machine when capron and textolite bushings are used. The greatest reduction in noise level is observed at frequencies which are multiples of the primary rotating frequency of the cutting shaft. /56

It was established in these experiments that the reduction in noise level depends on the type of bushing seating. Thus, when a hot or press fit of the bushing on the outer bearing ring was used, the noise reduction decreases sharply. A similar phenomenon was observed with a transition fit and particularly with a clearance fit.

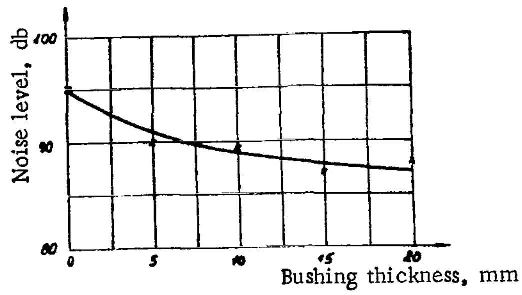

Figure 2. Noise Level of SF4-4 Machine as a Function of Vibration Damping Bushing Thickness.

The results produced can be explained easily if we consider that the attenuation of oscillations in bearing units with vibration dampers occurs due to deformation of the bushing material in thickness. When fits are used with considerable interference, the capability of capron for this type of deformation drops sharply. For this reason, noise reduction is practically zero. With fits closer to clearance fits, the contact between the outer ring of the bearing and the bushing is decreased, while with clearance fits it becomes minimal. The presence of the clearing causes intensive mechanical noise. The significance of the seating of the bushing as to its external diameter is less significant when the thickness of the bushing (capron) is changed. Analysis of the graph indicates that the optimal bushing thickness is 10-15 mm.

Thus, with proper selection of material, geometry and type of fit of the bushing around the outer bearing ring, vibration-damping bushings can provide a significant reduction in noise level.

It must be noted here that this can be done using practically any woodworking or metalworking tool.

INVESTIGATION OF VIBRATIONS WITH RANDOM AMPLITUDES AND FREQUENCIES

G. I. Anikeyev (Moscow)

An analysis is presented of the parameters of a random process at the output of the system characterized by its frequency response as function of the parameters of the input process and the frequency response of the system.

The characteristics of the random process at the output of the measuring system are used to determine the parameters of the process at its input. Four equations are produced, from which the mathematical expectation and variance of the amplitude and phase of the process at the input are determined. As an example, results are presented from the investigation of pulsations of pressure in the flow-through portion of a powerful hydraulic turbine.

SOME SPECIFICS OF THE INTERACTION OF DYNAMIC SYSTEMS WITH DISTRIBUTED AND LUMPED PARAMETERS

L. V. Goykhman and V. S. Naumenko (Moscow)

The interaction of a system with lumped parameters (railroad train) and a system with randomly distributed parameters (railroad track) is studied.

It is demonstrated that the system with randomly distributed parameters, as a result of nonlinearity of elastic and damping characteristics, is a set of oscillators with unstable frequencies.

The possibility of formation of traveling waves in a discretely anisotropic medium is studied.

The presence of wave processes in the track is detected and a model of formation of traveling and standing waves is suggested. The rails and ties must be looked upon in this case as a basic wave guide.

The possibility is studied of converting the system with distributed parameters to a system of lumped parameters. The lumped parameters selected are the mean or maximum parameters of the track (depending on the distribution law).

Then, a single track-train system is studied. The natural coupled frequencies of this system will be random.

A method is suggested allowing the instantaneous values of parameters in /58
the system without oscillation to be determined. The number of independent parameters determined in each of the cross sections of the track is equal to the number of degrees of freedom. The essence of the method consists in feeding a perturbation of the "white noise" type to one of the elements of the system in a predetermined frequency band, with a simultaneous Fourier transform of the signal at the output of any element. The spectrum produced is a signal of the frequency characteristic of the system and consequently will have maxima at the natural oscillating frequencies. Expanding the frequency determinant of the system, we produce n algebraic equations of order 2n for the frequency with n unknown parameters in each cross section. The number of cross sections is determined considering the least oscillating frequency of the para-

meters, while the distance between cross sections is determined using the Nyquist method.

For the masses, natural oscillating frequency and rigidity parameters determined, the distribution and their numerical characteristics are determined, and correlation and spectral analysis are performed. Using the equations of motion of the system and the expressions for unknown parameters produced, it is easy to go over to determination of the statistically averaged attenuation coefficients of the system.

A SET OF APPARATUS FOR MEASUREMENT OF OSCILLATIONS OF A MOVING TAPE

A. A. Alekna, P. A. Varanauskas, Z. F. Dontsu, V. T. Kolishchuk, K. M. Ragul'skis, M. P. Sukharev, Ye. N. Travnikov, A. V. Chepulkauskas (Kaunas)

This work presents a theoretical foundation, plus planning and experimental data for a set of apparatus for measurement of the transverse oscillations of a moving magnetic tape with uneven edges. The complex of equipment was developed at the laboratory of vibration studies of Kaunas Polytechnical Institute.

A number of works have been published in the domestic and foreign technical and patent literature on the measurement of transverse oscillations of a moving magnetic tape in a tape drive mechanism. Some works claim complete elimination of error resulting from uneven edges of an unevenly moving tape during measurement of transverse oscillations by a contactless method using sensing elements and an optical-mechanical system. It has been proved at the vibration studies laboratory recently that these errors cannot be com- /59 pletely eliminated, regardless of the number of stationary sensors used. Increasing the number of sensors decreases the influence of unevenness of tape edges on tape movement measurement. Other methods have been sought, reducing the error on the basis of the conditions of compactness of individual devices in the set of apparatus. Unevennesses with long repetition period, resulting from technological distortions of the tape during cutting, are random in nature, although they are near sinusoidal. Two sensors can be used, placed at a separation distance of one fourth the measured wave length. During motion, one sensor is uncovered, while the other is covered and the sum of the distortions produced will be less than the distortions from a single sensor. It is established that if the wave length has considerable deviation from the nominal (about 25%), the inaccuracies in measurement resulting from unevenness of tape edge will be over 30%. A similar phenomenon is produced from a lack of correspondence between the distance between sensors and wave length. With complete correspondence of these quantities, errors still result, since one sensor is uncovered with a delay of a quarter period of the wave being studied. This is eliminated as the distance between sensors is equal to one half wave length. As a result, precise selection of distance using this method of measurement decreases the errors.

STUDY OF VIBRATION OF ROTATING DISKS WITH MAGNETIC COATINGS

Yu. Yu. Getsevichyus, I.-M. P. Pakaushite, K. M. Ragul'skis (Kaunas)

Magnetic disks are used in disk memory devices (DMD). DMD have high capacity, approaching that of magnetic tape storage units, and rapid selection of the information required, comparable with that of drum memory units. These significant advantages of DMD have resulted in their wide and ever increasing application in modern computers and for a number of special purposes.

In magnetic storage, one of the significant problems in determining its quality is the problem of providing a constant operating clearance between the information carrier and the magnetic head in case of contactless recording and the provision of good quality contact between the information carrier and head in case of contact magnetic recording. A constant working clearance or good quality contact between carrier and head can be provided by controlling transverse oscillations of the rotating disk and the dynamic characteristics of the /60
head suspension system.

Methods of statistical dynamics, particularly correlation methods based on the study of the relationship between certain characteristics of input and output processes are used to study the transverse oscillations of a thin rotating disk of constant thickness, since the behavior of the disk depends on a number of random factors. These include the geometric and physical parameters of the disk itself: random deviations from ideal geometric form, variation in boundary conditions, fluctuations in elastic and strength characteristics of the material, etc. The transverse oscillations of a thin rotating disk were looked upon as a random stable process, satisfying the ergodic hypothesis. The realizations of the process were produced experimentally using contactless capacitive sensors with the corresponding amplifying apparatus. Investigation of transverse oscillations of a rotating disk with magnetic coating were performed using two plans. According to the first plan, the oscillations of the disk were received by contactless sensors, the signals from which entered the amplifying apparatus and were further fed to the recording apparatus. The recorded oscillations of the disks, in the form of curves, were converted to discrete form and transmitted to a computer for statistical processing. It was found that this plan of investigation of oscillations is quite ineffective. It is cumbersome and inaccurate, since the oscillations of such a precision device as a DMD are recorded by an inaccurate method, which is true of oscillographs used for the recording of dynamic processes. In these oscillographs, the drive of the tape on which the process is recorded is not highly even. Significant errors are also introduced during the process of conversion of the curves of oscillations to discrete form.

It is considerably more effective to utilize another method of investigation of disk oscillations, as we have. The analog signal from the amplifier apparatus is sent to a special converter used with the Minsk-22 computer. The eight channel converter, the speed of one channel of which slightly exceeds the speed of the main memory of the computer, records the indications of the sensors at evenly spaced time intervals and transmits them to the computer memory in the form of discrete pulses. This plan was used to calculate estimates of probability characteristics (correlation functions, spectral densities, histograms of amplitudes of transverse oscillations) of the disk-hub-clamp system and disk-contact head retainer system, as well as the dynamic characteristics in the form of estimates of the transfer functions and frequency characteristics. /61 The relationship of disk oscillations to its macro characteristic surface is also determined, the propagation of the level of oscillations along the radius of the disk is established, and the principal factors causing disk oscillations are determined.

One satisfactory method is the method of recording oscillations on magnetic tape using high-precision tape drive mechanisms, followed by conversion and input of data on the process to computer memory. This method of input of experimental data to computer memory allows eliminating a number of mechanical links, the unevenness and delay in motion of which introduce significant errors.

SOME PROBLEMS OF PRACTICAL ANALYSIS OF VIBRATION-ACOUSTICAL PROPERTIES OF CENTRIFUGAL PUMPS

D. V. Grokhovskiy and V. M. Rogachev (Leningrad)

The difficulty of the problem of determining the causes of increased structural noise level in pumps in various frequency ranges results from the continuity of the frequency spectrum of vibrations and the dual nature of the information included in the spectrograms. The variety of factors influencing the vibration-acoustical characteristics of the pump and the absence or in many cases the impossibility of establishing the functional relationship between parameters of the oscillating process and the design of the machine do not allow a theory of vibration-acoustical diagnosis of pumps to be constructed at the present time.

Works are described, performed during vibration-acoustical diagnosis of one pump, and calculation and experimental results are presented, in which the physical nature of the perturbing forces in various portions of the frequency range are studied, the specifics of the design and its dynamic modeling in these frequency subranges are presented, and the statement of expedient experiments is analyzed.

DETERMINATION OF PRIMARY SOURCES OF NOISE IN MACHINES BY CORRELATION METHODS

G. A. Leont'yev and P. M. Shul'ga (Volgograd)

A correlation method is used to determine the noisiest machine in one shop of an oxygen plant and reveal the interrelationships of noises created by the machine and vibrations of its individual units and parts. Autocorrelation and mutual correlation functions of the vibrations of individual units and the noise of a silk spinning machine were produced experimentally. The autocorrelation functions of the vibrations of the spindle, guide roller and reducing gear body were used to calculate the spectral densities. Analysis of mutual correlation functions of vibration and noise allowed the noisiest unit of the machine to be indicated. /62

The autocorrelation and mutual correlation functions were produced using a common apparatus: a noise meter, vibration measuring installation and dual-beam electronic oscilloscope with a photographic attachment. The oscillograms of the processes being studied were tabulated and fed into the Nairi digital computer for calculation of statistical characteristics.

STUDY OF OPTICAL DYNAMIC MODELS OF FLEXIBLE ROTOR SYSTEMS BY THE LP-SEARCH METHODS

M. F. Zeytman and R. B. Statnikov (Moscow)

Cybernetic diagnosis of dynamic models of rotors is based on the new LP-search method, which determines optimal versions of models in the space of variable parameters. Investigation of multivariate dependences is based on the use of the Haar function.

It is suggested that points $Q_1,\ldots,Q_n$ of an LP_τ-sequence, calculated on the basis of the guiding points $V_1\ldots V_s\ldots$ be used as pseudorandom points in an n-dimensional cube. The P_τ networks guarantee optimal order of convergence in the space of parameters of the functions. Calculation of these points is based only on logical operations, and their calculation by computer is quite simple. In this work, LP-search was used to study flexural oscillations of a flexible /63 vertical rotor, oscillating in the field of the force of gravity. Free and forced oscillations of the rotor are studied and optimal values of parameters satisfying the established quality criteria are sought.

STUDY OF METHODS OF SEPARATING USEFUL SIGNALS FROM GENERAL NOISE OF DUPLICATING MACHINES IN SOLVING THE PROBLEM OF AUTOMATIC RECOGNITION OF ORIGIN OF NOISES IN CONCRETE MECHANISMS USING DIGITAL COMPUTERS

O. K. Postnikov (Moscow)

The complexity of the kinematics of duplicating machines makes separation of noise sources difficult.

This problem can be solved during duplicating production by non-separated search and determination of sources.

A method is suggested for machine investigation of duplicating machines for noise with automatic recognition of signals produced. In creating the method, the specifics of duplicating machines, including their high kinematic complexity and rigidly programmed operating cycle, were taken into consideration.

Methods developed allow information to be extracted from a realization X(t) not only concerning the noise level of any given mechanism, but also allowing estimation of the quality of manufacture and installation of individual parts and units of the machine. When the proper statistics are selected and particular investigations are performed, this method can be promising for use in diagnosis as well.

During preparation of a problem for computer solution, the realization of the signal X(t) must be performed in advance with precise determination of the rotating speed of the main shaft. The use of position or angle markers of the mechanisms being studied is obligatory. Preparation of the problem also includes composition of a data block of kinematic dependences of all mechanisms of the machine, and where pulse noises are present, generalized dynamic calculations as well. (The algorithm for composition of the data block and calculation of dynamic parameters required for investigation allows automation of this calculation).

The duration of recording of realization X(t) and the frequency of the reference signal are determined by the researcher on the basis of rules which are developed.

An analog to digital converter is used to transmit the information to a digital computer. The operating mode of the converter is normalized by the recommendations used for the investigation. /64

Using an algorithm which is developed, after the necessary information has been fed into the computer, the experimental data are processed in the machine.

If the mechanisms of the machine being studied have frequencies which are identical or multiples of each other, several oscillation sensors must be used.

After the "useful" periodic signals and pulse components have been separated, the "random" periodic components are analyzed for purposes of recognition.

An experimental check of the method performed using the POL-6 two-color offset machine, gave positive results.

STATISTICAL STUDIES OF RANDOM VIBRATIONS OF A RAILROAD CAR BODY

G. M. Frolov (Moscow)

1. In order to estimate the dynamic qualities of moving elements, the oscillations of various elements and structures of cars operating under intensive vibration conditions are studied. The vibrations of a car body consist of a superposition of a large number of harmonic oscillations with random amplitudes and frequencies. Generally they can be placed in the class of complex random vibrations.

2. A combination of electronic equipment has been developed for automatic construction of the bivariate distributions of probability densities of the maxima and durations of complex oscillations of the process being studied. The measurements are performed in the time between two neighboring intersections of the zero level by the signal. This method of analysis of oscillations is used, for example, in determining the smoothness indicators of the cars using the Sperling method.

3. The set of apparatus consists of specially developed and standard electronic devices, including a magnetic recorder for the vibrations being studied and a special digital pulse analyzer. The measured parameters of oscillations--maxima and duration--are converted to pulse trains. In correspondence with the number of pulses in the trains, the pulse analyzer sorts oscillations as a function of their parameters and records this information in memory. The results of processing of long realizations of the process are output as correlation tables, fully defining the one-dimensional distributions of each of the /65
parameters studied and allowing the stochastic dependences between them to be studied. These correlation tables are input to a universal digital computer for calculation of the smoothness indicators (by processing of vertical and horizontal accelerations of the car body), statistical characteristics of the one-dimensional probability density distributions and correlation factors of the process parameters studied.

4. Investigation of the probability relationships between the vibrations of any two structural elements, for example between the vibrations of the body and trucks of a railroad car, is the most difficult form of analysis. In this case, when the mutual correlation between the instantaneous values of the two

processes is studied, the signals must be quantized with respect to time and the instantaneous values of signals converted to pulse trains as described above. Correlation tables of two processes can be used not only to calculate the correlation coefficients, but also to determine various statistical coefficients of the ratios of the two random vibrations. These coefficients are the tangents of the inclination angles of the regression lines of the processes being studied.

5. All calculations in processing of the vibration data are performed by electronic digital computer, allowing the process of investigation of vibrations to be automated.

METHOD OF MEASUREMENTS DURING INVESTIGATION OF MOTORCYCLE NOISE

Yu. V. Skoryukin, A. A. Strokin, A. S. Terekhin, V. V. Tupov and Ye. Ya. Yudin (Moscow)

At the present time, there is no satisfactory theory for calculation of the noise of a motorcycle and its muffler system. Therefore, various methods of experimental study are of considerable significance. One such method is the subject of the present work.

As we know, the principal components in the noise of a motorcycle are the exhaust noise, intake noise and noise of the engine itself. Therefore, tuning /66 of the intake and exhaust muffler system has a strong influence on the power and economy figures of the engine. Under plant conditions, measurement of these indicators generally is performed on stable, specially constructed test stands. Therefore, the necessity frequently arises of using these stands also for measurement of the noise characteristics of engines. We suggest that the reflected field method be used for this purpose, since it corresponds most closely to natural conditions.

In order to confirm the correctness of this method, we studied the acoustical properties of test stands at the V. A. Degtyarev Plant.

For this same purpose, we performed comparative measurements of the sound power of a motorcycle engine in a free sound field by the reflected sound field method, showing good convergence of the two methods. Furthermore, the results of measurements under test stand conditions were tested experimentally by road tests of motorcycles. To do this, the test stand measurement data were used to calculate the expected sound pressure spectrum under road conditions. The road test measurements showed satisfactory correspondence of calculated and experimental data.

Based on the studies performed, we can consider test stand acoustical measurements reliable. This allows us to expand significantly the experimental capabilities of test stands of this type, available in various organizations, fitting them simultaneously for measurement not only of power but of acoustical indices of engines.

DETERMINATION AND CALCULATION OF THE CHARACTERISTICS OF NOISE CREATED BY TURBO-PROP PASSENGER AIRCRAFT IN THE AREA OF AIRPORTS

V. S. Okorokov (Moscow)

A method is suggested for determining the maximum noise level of aircraft, both when parked and when in flight.

The initial parameters used are the power applied to the propeller, rate of propeller rotation, diameter and the number of blades, flight altitude and speed and the arrangement of the power plant. The formulas and nomograms suggested allow the noise of aircraft in operation and newly designed aircraft with turboprop engines to be calculated with accuracy sufficient for practical purposes. Estimation of noise is performed in effective perceived noise levels in correspondence with the recommendations of the International Standards /67
Organization and the plan for the corresponding domestic All-Union State Standard.

The results of experimental studies of noise characteristics of domestic aircraft tested parked under various operating conditions agree well with the calculated values.

STUDY OF AIR NOISE OF PNEUMATIC LOOM BY STATISTICAL METHODS

L. P. Bastite, A. Yu. Klyuchininkas, V. K. Naynis, K. M. Ragul'skis (Kaunas)

A comparison of spectrograms measured during the operation of a shuttle loom of any type and a pneumatic loom shows that the noise created by pneumatic looms consists of high frequency aerodynamic and turbulence noise. In the octave 1, 2, 4 and 8 KHz bands, the noise level exceeds the norms by 2, 2, 4 and 14 db respectively.

The principal sources of high frequency noise in a pneumatic loom are the units supplying compressed air and drawing away the weft. Figure 1a and b shows an oscillogram of the noise of the air supply unit. The duration of an /68
air blast is 96 angular degrees of rotation of the main shaft of the machine.

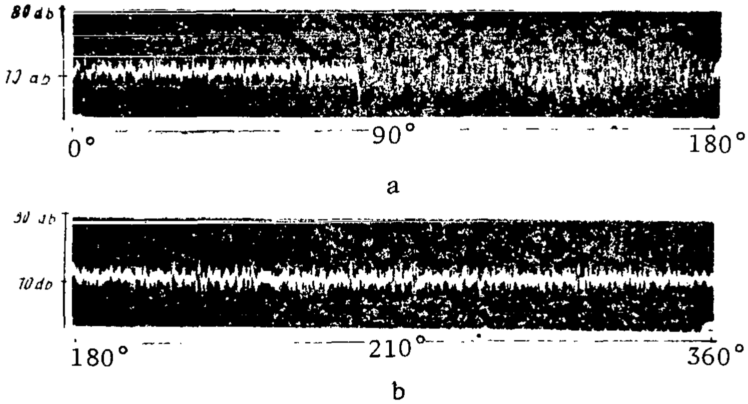

Figure 1. Oscillogram of Noise of Air Feed Unit: a, 180° rotation of main shaft; b, 180-360° rotation of main shaft; time mark represents 500 Hz.

Pulsations in air pressure far from the zone of the air stream and in it are studied. Since the airflow field in the stream is random, a statistical description of the phenomena is presented. The relationship between pressure pulsations in the airstream and the narrow band noise spectra is determined.

An experimental study of the pulsations of air pressure in the stream was performed using a multichannel recording apparatus. The data of experimental studies, produced in the form of oscillograms, are processed by statistical methods on a digital computer.

A WATER SPRAY AS A SOURCE OF RANDOM FORCE FOR ACOUSTICAL MEASUREMENTS

T. F. Demidenko, N. V. Stepanova, V. I. Shmal'gauzen (Moscow)

In order to estimate the effectiveness of the noise and vibration insulation of various devices, it is desirable to have a point source of noise action, with a broad spectrum, stable and easily reproducible effect. D. G. Tonkonogov has suggested that a stream of liquid be used for this purpose, spraying at 2-4 atm pressure from a small aperture (0.3-0.6 mm in diameter). This report presents the results of studies of the spectral properties of the noise excited by a broken stream of liquid.

We know that a stream of water is unstable in air and at some distance from the nozzle is broken into individual drops, the mean radius of which is approximately equal to the diameter of the aperture: $R \approx d$ (see [1,2]). The formation of drops is a random process and therefore when the broken stream strikes a solid surface, a random sequence of pulses of force arises. The mean pulse repetition frequency ν can be determined from the law of conservation of mass. If M is the flow rate of water per unit time, v is the stream velocity, then

$$\nu = \frac{M}{\frac{4}{3}\pi \bar{R}^3 \rho} = \frac{3}{16}\frac{d^2 v}{\bar{R}^3} \simeq \frac{3}{16}\frac{v}{\bar{R}}. \qquad (1)$$

In order to determine the spectral density of the random force, it is necessary /69
to know the pulse form. Since it has been noted that the pulse has a steep leading edge and a more gradual trailing edge, the following empirical formula can be used to approximate the pulse form:

$$F(t) = C(v,\ R)\, e^{-\frac{t}{\beta T}} \left(\frac{t}{\beta T}\right)^{\alpha-1} \qquad (2)$$

Here $T = \bar{R}/v$ is the effective pulse length, α and β are parameters which do not change when $\bar{R}$ or ν change, amplitude $C(\nu, \bar{R})$ is determined from the law of conservation of momentum:

$$C = \frac{4\pi \bar{R}^3 \rho v}{3\beta T \Gamma(\alpha)}$$

The spectrum of an individual pulse of fixed form will be /70

$$Q(\omega) = \int_0^\infty F(t)\, n^{i\omega t}\, dt = 4\pi \bar{R}^3 \rho v (1 - i\omega\beta T)^{-\alpha}. \qquad (3)$$

For a random sequence of such pulses, following at mean frequency ν, the spectral density is determined by the expression

$$S(\omega)=\nu\,[Q(\omega)\,Q^*(\omega)]=\frac{\pi^2}{3}R^6 v^3 \rho^2\,[1+(\omega\beta T)^2]^{-\alpha} \qquad (4)$$

Figure 1 shows the results of an experimental determination of the spectral density of the noise of the stream. Measurements were performed using a piezoelectric receiver, the sensitivity of which in the frequency area studied (10-100 KHz) can be considered constant. The points relating to various aperture diameters and various stream velocities fall near the curve constructed using formula (4) with the values of the parameters $\alpha = 1{,}2$; $\beta = 1$. Thus, where $\omega \gg V/\bar{R}$, the spectral density drops off as frequency increases as $\omega^{-3.4}$.

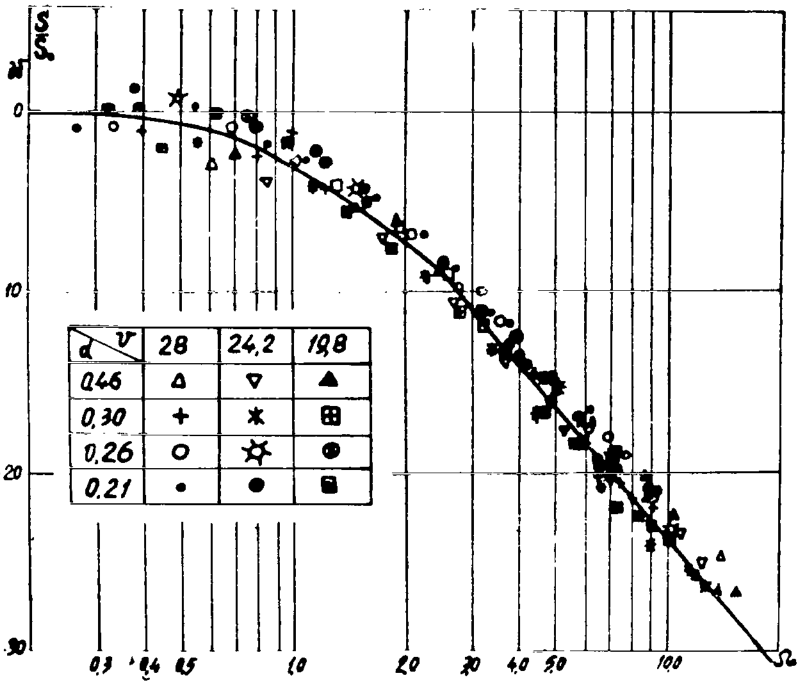

Figure 1.

Thus, a broken stream of liquid, flowing from a small aperture, is a practical point source of random force with known properties.

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NOISE AT ACOUSTICAL RECEIVER RESULTING FROM SPATIALLY NONCORRELATED SOURCES DISTRIBUTED OVER A SURFACE

V. B. Kobel'kov and D. G. Tonkonogov (Moscow)

It is suggested that the acoustical properties of a device and receiver be characterized by the response function H(M,f), equal to the voltage at the output of the receiver upon excitation of point M of the surface at frequency f by a variable force with unit amplitude. This function can be experimentally measured. The nature of the sources can be described by a function of the /71 surface density of exciting forces $\sigma(M,f)$, equal to the mean amplitude of the force at frequency f per unit surface area, including point M. Assuming the sources on surface S to be noncorrelated, we calculate the resulting noise at frequency f as:

$$U(f)=\sqrt{\int_S H^2(M,f)\cdot\sigma^2(M,f)\,dS}. \tag{1}$$

As an example, let us analyze a disk with receivers set in the center, for which functions H and σ have the form shown on Figure 1. The results of calculations of the noise level for this case using formula (1) are presented on Figure 2.

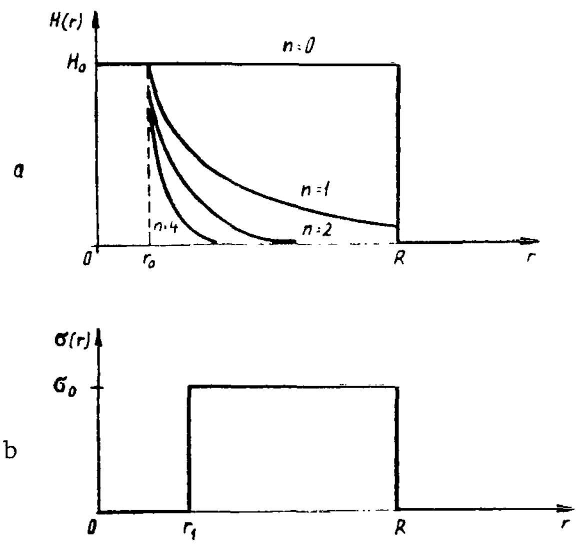

Figure 1. Form of Functions: a, Response; b, Excitation density for circular model of radius R with receiver of radius v_0 set in center.

where $r_1 = 0$ (receiver set in wall of device, surface of which is excited evenly) for $n > 2$, the noise level remains practically unchanged, which can be looked upon as a criterion of correctness for measurements of function σ, or as the corresponding limit in attempts to decrease the noise on the receiver, resulting from the surface areas surrounding the receiver. If the area of excitation is remote from the receiver ($r_1 > r_0$), the noise level depends strongly on the value of n, this dependence being steeper, the greater r_1, which can be used to determine means of protection from noise or for determination of r_1 on the basis of results of measurement of noise at various n. For case of axial symmetry with a circular boundary of excitation at distance r along the generatrix (if, where $r > r_1$, function $\sigma(r,f)$ does not increase to compensate for the decrease in function $H(r,f)$) for various functions $H(r,f)$, the following equality obtains:

$$U(f) \simeq Const \cdot H(r_1 f) \qquad (2)$$

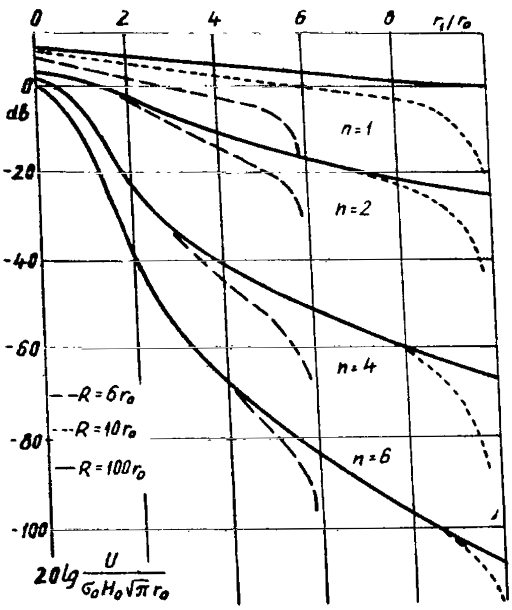

Figure 2. Noise Level as a Function of Position of Excitation Boundary r_1 with Various Exponential Decreases in Response Function n and Dimensions of Disk R.

In other words, for this case such characteristics as $H(0,f)$; n; dimensions and form of areas of the surface remote from the receiver have no significance. It follows from (2) that if for a certain device the noise $U_1(f)$ and the response function are known with an accuracy equal to coefficient $\tilde{H}_1(r_1,f)$ for another device with the same shape and nature of excitation the noise level is defined as: /73

$$U(f) = U_1(f) \cdot \frac{\tilde{H}(r_1, f)}{\tilde{H}_1(r_1, f)} \qquad (3)$$

where $\tilde{H}(r_1,f)$ and $\tilde{H}_1(r_1,f)$ are measured under identical conditions without absolute calibration of the exciting forces.

STUDY OF THE NOISE OF A DIESEL ON A TRACTOR

Yu. Deykus, V. Lukanin, V. Efros (Kaunas)

Investigations have shown that the noise in the operator's seat of a tractor is radiated by the panels of the cab, excited by forces developed as the engine operates, and also as a result of propagation of sound energy through the structures of the tractor.

Studies were performed using type MTZ-50 tractors, equipped with water cooled (D-50) and air cooled (D-37Ye) engines. The specifics of arrangement of the engine on the tractor include rigid mounting on motor mounts and flange connection to the transmission of the tractor.

Vibrations can be decreased by making structural changes to the method of connection of the engine to the transmission.

Figure 1 shows the results of measurement of vibrations at a point on the floor of the cab. Spectrum 1 was produced with standard connection of the engine to the transmission, spectrum 2--with the engine disconnected, spectrum 3--shows vibrations of the engine itself.

The noise in the operator's seat was 104 db and 98 db respectively. The engine in all cases operated at 1800 rpm without load.

This figure shows the expediency of reducing the vibration propagating from the engine to the cab.

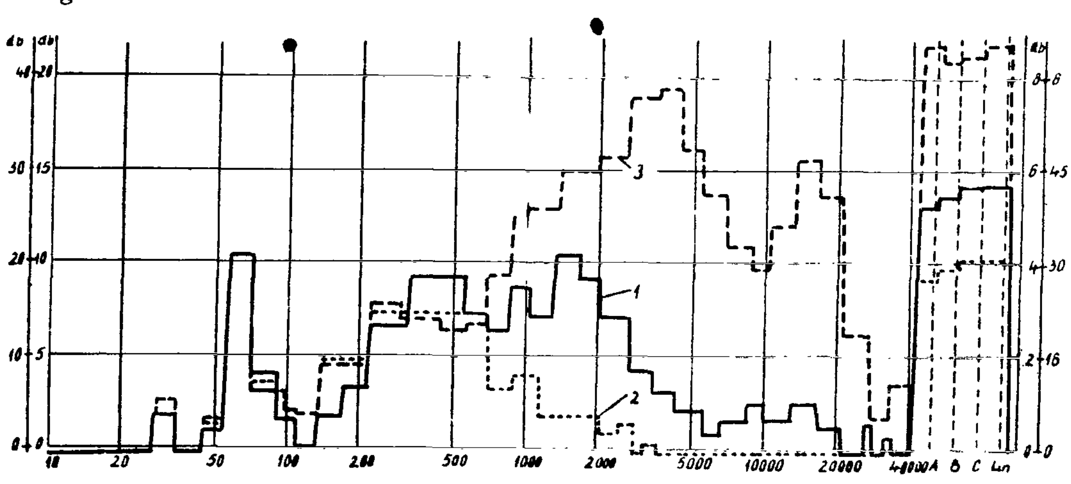

Figure 1.

Comparison of noise in the cab of the tractor using engines with various types of cooling shows that the air-cooled engine causes a noise level of 101 db, while the water-cooled engine causes a noise level of 108 db. The comparison was made at 1750 rpm, without load.

AUTOMATIC METHOD OF STUDYING SOUND INSULATION OF CYLINDRICAL TUBES WITH VARIOUS SOUND RADIATION CONDITIONS

D. R. Guzhas (Vil'nyus)

The study of the sound insulation (SI) of cylindrical tubes is becoming a pressing problem in the area of production noise control. In gas supply systems, as strongly turbulent gas streams flow through tubes, intensive noise arises, reaching 110-120 db outside the tube. /75

The installation developed allows the SI of tubes up to 10 m in length with diameters up to 1 m to be studied with sound sources both outside and inside the tube (Figure 1).

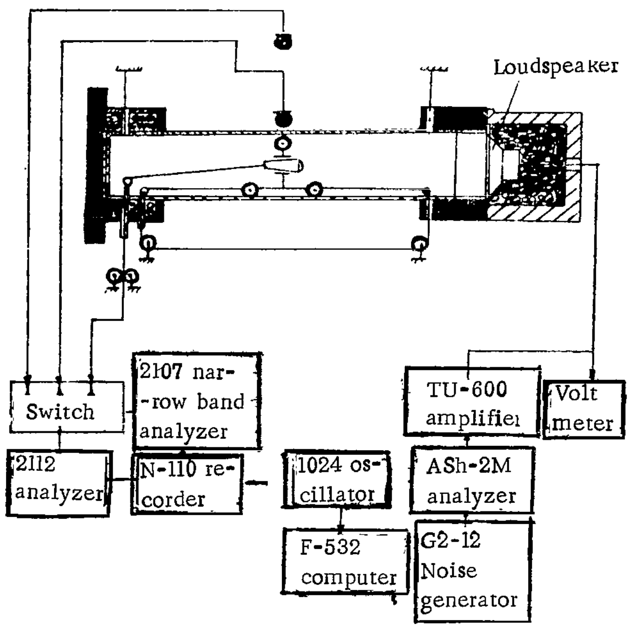

Figure 1.

The installation includes guide tubes installed outside the tube, on which special roller collars are seated. The collars carry a supporting ring for six microphones located at angles of 60° around the tube. The microphone ring is moved automatically along the tube with a strip chart recorder and a special drive system. The microphone within the tube can be moved using a line along the axis of the tube. The unevenness of the sound field through the cross section of the tube is determined by movement of the microphone over the radius of the tube using a flexible connector mounted to the microphone /76

support truck. This truck is moved automatically by the recorder. A measuring /77
rod determines the position of the microphones inside and outside the tube.

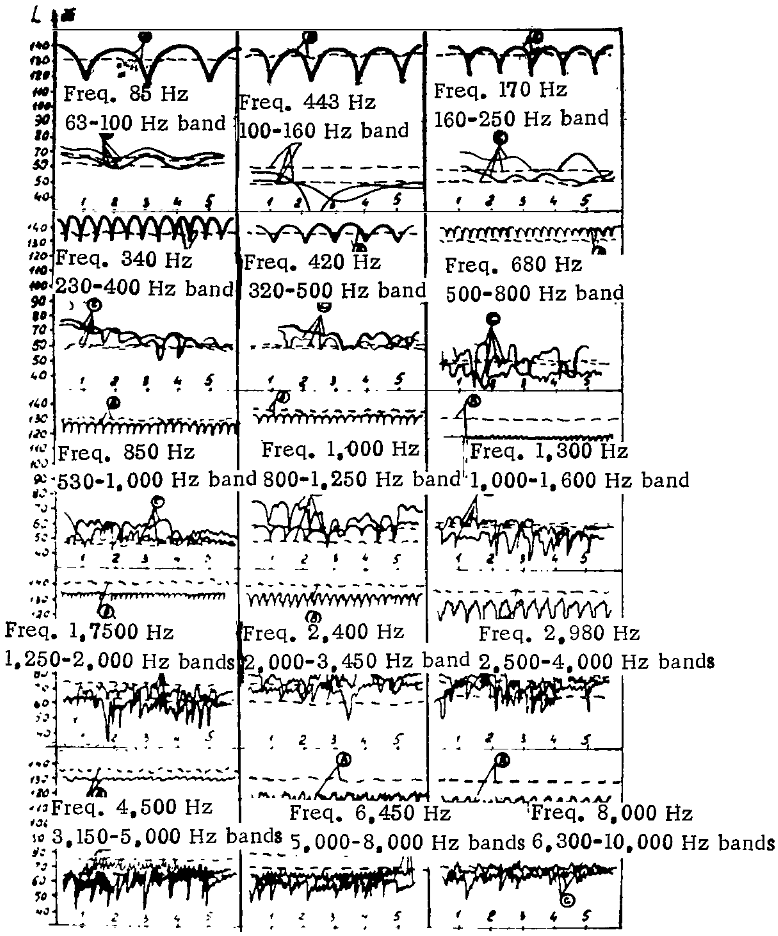

Figure 2.

The electroacoustical portion of the installation includes master and measuring-recording channels (Figure 1). A type 50 GRD-8 horn loud-speaker is used to supply the sound pressure. The master channel allows sound sources to be excited with sine wave tones or one-third octave bands of white noise.

A type N-110 strip chart recorder is used to record the level of sound pressure along the tube at the frequency measured. The measuring micro-phones outside are placed right at the surface of the tube (distance 0.5-1.0 cm) and at a distance of 50 cm from the tube wall.

The authors studied the SI of two cylindrical tubes 6 m in length and 219 mm in diameter with various wall thicknesses: b_1 = 8 mm and b_2 = 2 mm using this device. With the sound source inside the tube, the internal and external sound fields were measured. The measured level of sound pressure along the tube with b_1 = 8 mm when excited by bands of white noise and sine wave tones is shown on Figure 2. The solid curves show the results from the sine wave tones, the dotted curves show the results from the white noise bands, the heavy curves were measured at 0.5 m, the fine curves--at 5-10 mm; curves c show results of measurements outside, curves b show results of measurements inside.

The sound insulation of cylindrical tubes decreases with increasing frequency by an average of 5-6 db per octave, i.e., the frequency tendency of the sound insulation of shells is opposite to the tendency of the sound insulation of plates.

SIGNIFICANCE OF INFORMATION ON CERTAINTY OF UNSTABLE NOISE IN THE PROCESS OF ITS ACTION ON THE HUMAN ORGANISM

G. A. Suvorov and A. M. Likhnitskiy (Leningrad)

It is suggested that an information description of unstable noise be added to the power evaluation (mean power) and spectral evaluation (time-averaged spectrum), using for this purpose the statistical envelope of the process:

$$P_{ms}(t) = \sqrt{\frac{1}{T}\int_0^T h(t')\,|P(t-t')|^2\,dt'}$$

where $P_{ms}(t)$ is the statistical envelope of the process, P(t) is the instantaneous sound pressure, h(t) is a weight function. Function h(t) describes the time window within which the human ear is insensitive to the phase structure of a signal. The effective width of the smoothing interval is assumed to be $\Delta t_0 \simeq 10$ msec. /78

The effect of noise with a complex envelope is analyzed as the result of interaction of the organism with a stimulus which changes with time; as the result of this interaction, the organism develops a strategy to minimize the biological effect of the noise (so-called dynamic adaptation), the least influence of an unstable stimulus being noted when its parameters are known in advance. It is assumed that the human organism partially solves the difficulties resulting from a delay in its protective reaction to sudden effects by extrapolating future moments of appearance of a stimulus on the basis of information concerning the envelope of the noise process in the past. The minimum attainable uncertainty can be calculated on the basis of the probability characteristics of the noise envelope. It is assumed that the entropy is used as the measure of uncertainty.

In order to determine the most characteristic properties of dynamic adaptation, we studied the effects of unstable noise, an aperiodic sequence of short rectangular white noise pulses t_u in length. In the first approximation, this sequence follows Poisson's rule. The uncertainty of the moments of appearance of the pulses required that the test subjects perform probability extrapolation on the basis of information on the preceding intervals between the initial moments of pulses (waiting intervals $T^{(i)}$). Since the intervals

$T^{(i)}$ are statistically independent and are the only source of information on the stimulus in the "future," the uncertainty is determined by the probability density of the Poisson process:

$$w(T^{(i)}) = \frac{1}{\bar{T}^{(i)}} e^{-\frac{T^{(i)}}{\bar{T}^{(i)}}} \qquad (T^{(i)} \geqslant 0)$$

$T^{(i)}$ is the mean expectation interval, then the entropy is expressed as

$$H[W(T^{(i)})] = -\int_0^\infty \frac{e^{-\frac{T^{(i)}}{\bar{T}^{(i)}}}}{\bar{T}^{(i)}} \log \frac{e^{-\frac{T^{(i)}}{\bar{T}^{(i)}}}}{\bar{T}^{(i)}} \, dT^{(i)} - \log \Delta t_0$$

Dimensionality $I/\Delta t_0$ represents the permissible uncertainty of the expectation interval for the ear, resulting from the integral properties of the ear. /79
After integration and introduction of corrections for pulse overlap, the entropy is expressed by

$$H(T^{(i)}) \approx \log \frac{e(\bar{T}^{(i)} - t_u)}{\Delta t_0} \qquad (t_u < T^{(i)})$$

The study of the specifics of the effects of noise as a function of the distribution of sequences of pulses of equal mean power in time showed that the most sharply expressed biological effect is that of aperiodic noise, characterized by the maximum uncertainty of expectation intervals between pulses, the influence of the noise increasing with increasing entropy.

THE PROBLEM OF THE STATIC CALCULATION OF VIBRATION INSULATION SYSTEMS WITH SIX AND TWELVE DEGREES OF FREEDOM

Yu. P. Busarov (Vladimir)

Various systems of equations are concluded for static calculation of the vibration insulation systems with six and twelve degrees of freedom. The following goals can be achieved by solving these equations:

1. Calculation of linear displacements and angles of rotation of vibration-insulated object.
2. Calculation of deformations and reactions of shock absorbers.
3. Calculation of inserts and recesses beneath shock absorbers used for elimination of skew of vibration-insulated object.

These calculations can be performed for shock absorbers with linear and nonlinear static characteristics. In the latter case, the static characteristic is represented as a sum of a linear characteristic and a nonlinear remainder.

Solution of the equations involved in nonlinear static design of vibration insulation systems is performed using the method of successive approximations. The nonlinear remainders are discarded in calculation of the first approximation.

The work involves the apparatus of matrix calculus and n-dimensional vectors, which is convenient for static calculations by digital computer.

REDUCTION OF NOISE AND VIBRATION OF PNEUMATIC LOOM USING VIBRATION-INSULATING SUPPORTS

L. P. Bastite, P. I. Ilgakois, A. Yu Klyuchininkas (Kaunas)

One means of decreasing dynamic loads on the base and reducing the noise of looms is the use of vibration-insulating supports. New designs of vibration supports are suggested. /80

During a qualitative evaluation of vibration-insulating supports, the dynamic loads on the base and vibrations of the floor were measured during operation of a pneumatic loom installed on the second floor of a building.

Vibration-insulating supports are designed for the P-125 pneumatic loom. The effectiveness of application of vibration-insulating supports of the new design in order to decrease the effects of dynamic loads on the base and reduce the noise of the machine is demonstrated.

Results are presented from calculation of the reduction of noise level of the pneumatic loom when installed on vibration-insulating supports of the design suggested. Results are also presented from a calculation of the decrease in total noise power in a room containing 36 type P-125 machines when they are set on vibration-insulating supports of the designs suggested.

Experimental study of the dynamic loads on the machine and vibrations of the base is performed using a multichannel recording apparatus with seismic and piezoelectric sensors. The data on vibrations of machine and base are produced in the form of oscillograms and processed by statistical dynamics methods by digital computer.

GENERALIZATION OF THE HYPOTHESIS OF Ye. S. SOROKIN TO INCLUDE NONLINEAR ELASTIC DAMPING ELEMENTS

Yu. P. Busarov (Vladimir)

The hypothesis of Ye. S. Sorokin is extended to elastic damping elements with nonlinear elasticity curve. The generalization is performed by the method of direct linearization of the elasticity curve, allowing the equivalent rigidity $C(A,\omega)$ of an elastic element, generally depending on amplitude A and deformation frequency ω, to be introduced. This equivalent rigidity /81
is used in place of the constant rigidity in recording the absorption factor of an elastic element in its classical form:

$$\gamma(A, \omega) = \frac{S}{\pi C(A, \omega) A^2}, \tag{1}$$

where S is the area of the hysteresis loop.

In correspondence with this definition of the absorption factor, the generalized hypothesis of Ye. S. Sorokin is written in the form

$$N = -C(A, \omega)[1 + i\gamma(A, \omega)] \cdot \chi, \tag{2}$$

where N is the reaction of the elastic element;

χ is the deformation of the elastic element.

Based on the generalized hypothesis, the amplitude-frequency characteristic of the nonlinear oscillating system with mass m and one degree of freedom, perturbed by a harmonic force with amplitude F_0, is written in the convenient form

$$A = \frac{F_0}{\sqrt{[-m\omega^2 + C(A, \omega)]^2 + [\gamma(A, \omega) C(A, \omega)]^2}}, \tag{3}$$

known for the case of linear oscillations.

It is demonstrated that the solution of the nonlinear equation of an oscillating system

$$m\ddot{x} + N(x) = F_0 \sin \omega t, \tag{4}$$

where N(x) is the characteristic of the elastic damping element in the form of an experimental hysteresis loop, by the various approximate methods of

Galerkin-Ritz, the small parameter method, the harmonic balance method, the averaging method and others also leads in the first approximation to expression (3), although by a more complex means.

At the same time, in order to perform calculations using formula (3) it is sufficient to know only the dependence of rigidity S and absorption factor γ on the amplitude and frequency of deformation, which can be easily produced by processing of experimentally produced hysteresis loops.

OSCILLATIONS OF ELASTICALLY SUSPENDED BODY WITH CENTER OF GRAVITY MISMATCHED TO CENTER OF ELASTICITY OF SUPPORT

A. P. Zhurevskaya (Moscow)

The case of elastic placement of the body of an electronic apparatus on a platform which performs vertical oscillations according to a harmonic rule is studied. It is assumed that the center of gravity of the body not only does not correspond to the center of elasticity of the support, but is not located above it--which is rather frequently encountered in actual structures. The solution of the problem is performed in the primary coordinates, which are the angles of rotation of the body relative to the centers of the principal oscillations located with the static placement of the body in the same plane as the center of elasticity of the support. The deformations of elastic elements of the support of the body are determined. Conditions are studied under which "breakdown" of the suspension is possible. /82

ONE METHOD OF DAMPING PARAMETRIC OSCILLATIONS OF A ROD CONSIDERING A DAMPING SUSPENSION

D. Kh. Tsveniashvili (Moscow)

Problems of the dynamic stability of a rod subject to the influence of a longitudinal pulsating tracking load are studied when a supplementary mass (damper) is attached to it through an elastic element. The rod is represented as an elastic, massless element with a concentrated mass at its tip (as in the works of Ya. G. Panovko, et al). The vertical displacements of the mass of the damper and the concentrated mass of the rod are ignored. Thus, we produce a system with two degrees of freedom (horizontal displacements of the damper and the tip mass of the rod). An expression is composed for this two-mass system for the bending moment in an instantaneous cross section, after which, by using the Principle of d'Alembert, the following system of differential equations of motion is produced

$$\begin{gathered} \ddot{f}_1+\mu h_1\dot{f}_1-\mu h_1\dot{f}_2+(K^2\omega^2+\mu\Omega_1^2)f_1-\mu\Omega_1^2f_2=0, \\ \ddot{f}_2-h_1\dot{f}_1+h_1\dot{f}_1-\Omega_1^2f_1+\Omega_1^2f_2=0, \end{gathered} \tag{1}$$

where /83

$$\begin{gathered} \omega^2=\frac{(\alpha l)^3}{3(\sin\alpha l-\alpha l\cos\alpha l)}, \qquad K^2=\frac{3EI}{ml^3}, \qquad \Omega_1^2=\frac{C}{m_1}, \qquad \mu=\frac{m_1}{m} \\ h_1=\frac{q}{m_1}, \qquad \alpha=\sqrt{\frac{P(t)}{EI}}, \qquad p(t)=p_0+p_t\cos\Theta t, \end{gathered} \tag{2}$$

m is the concentrated mass of the rod, m_1 is the mass of the damper, C is the rigidity of the spring, $f_1(t)$ is the bending of the upper end of the rod, $f_2(t)$ is the deflection of the mass from the position of static equilibrium, K is the natural frequency of the rod without the damper, Ω_1 is the partial natural frequency of the damper, q is the damping factor, P_0 and P_t are constants, EI is the rigidity of the rod.

System (1) is non-Hamiltonian. Coefficient $\Omega_2(t) = \omega^2(t) = \omega^2(-t)$, i.e., an even function, while all remaining coefficients are constant quantities relative to time t. We assume that $\varepsilon = p_t/p_0 \ll 1$, i.e., we consider ε a small parameter. This corresponds physically to the case of slight modulation of the load. Expanding $\omega^2(p_0 \cdot p_*, \varepsilon)$ $(p_* = 20.19\ EI/l^2)$ in a series in power of ε, we have

$$\omega^2 = \omega_0^2\left(\frac{p_0}{p_*},\ 0\right) + \frac{\varepsilon}{1!}\dot{\omega}^2\left(\frac{p_0}{p_*},\ 0\right)\cos\Theta t + \frac{\varepsilon^2}{2!}\ddot{\omega}^2\left(\frac{p_0}{p_*},\ 0\right)\cos^2\Theta t + \ldots$$

Assuming $\tau = \Phi t/2$, $\lambda = 2/\Phi$, system (1) can be replaced by the following, equivalent system of first order differential equations:

$$\dot{f}_1 = \lambda f_3,\quad \dot{f}_2 = \lambda f_4,$$
$$\dot{f}_3 = -\lambda\,(K^2\omega_0^2 + \mu\omega_1^2) f_1 + \lambda\mu\Omega_1^2 f_2 - \lambda\mu h_1 f_3 + \lambda\mu h_1 f_4 - \varepsilon\lambda K^2\dot{\omega}_0^2\cos 2\tau f_1 + \ldots,$$
$$\dot{f}_4 = \lambda\Omega_1^2 f_1 - \lambda\Omega_1^2 f_2 + \lambda h_1 f_3 - \lambda h_1 f_4\,,$$

(considering that the damping factor is a small quantity).

The construction of areas of instability is performed using the method of I. G. Malkin, which is based on direct calculation of the values of characteristic indicators near the critical values of parameter λ. Analysis is performed for the first areas of instability of simple and combination parametric resonances. It is demonstrated that by attaching a dynamic damper to the rod, we produce, in place of a single area of simple parametric resonance for the case of a system without a damper, two areas of simple parametric resonance and one area of summary combination resonance. By proper selection of adjustment of the damper K/Ω_1, we can achieve a placement of areas of simple /84 and combination parametric resonances such that at what were the critical frequencies of the system without the damper, instability occurs only when a certain load level is achieved. Consideration of damping leads to rising of the areas of instability above the abscissa.

STUDY OF NOISE AND VIBRATION UPON IMPACT

Yu. D. Valanchauskas (Kaunas)

Pulse-type loads occur in the operating conditions of many modern machines and mechanisms.

This report analyzes a method for studying the free impact oscillations and noise characteristics of a rod device.

A special stand was constructed for performance of the experiments.

During the experimental studies, the force of impact, motion of colliding masses and noise were measured.

The data were recorded magnetically. Subsequent processing of results was performed using electronic computer equipment. The transfer function found between vibration and noise can be used in the diagnosis of machines and mechanisms.

THEORY OF MULTICHANNEL COMPENSATION SYSTEM FOR OSCILLATIONS IN STRUCTURE (FIELD) OF ARBITRARY FORM

B. D. Tartakovskiy (Moscow)

The possibilities of using radiating systems for compensation of oscillations at any number of points in a structure (field) of arbitrary form are studied. The compensating oscillations (field) are considered known, assigned from without or developing within the structure. Limitations of linearity are placed on the transmission of oscillations through the structure (on the transmission of waves in the medium) and on the conversion of signals in electromechanical channels and amplifiers. Forward and reverse design of the electromechanical compensation system are performed. In order to provide the required compensation of oscillations of the structure (sound field) at a certain (arbitrary) /85 number of points in the structure (sound field) at one or more frequencies, the necessary parameters of the electromechanical multichannel feedback system are studied. The data produced are used for determination of oscillations of the structure (sound field) outside the tested points in the mechanical system, providing for a fixed attenuation of oscillations at the selected test points, located at the limit or outside the structure. After calculation of parameters of the multichannel feedback system, the areas of compensation of oscillation and zones of their amplification are determined, as well as conditions limiting autoexcitation of the multichannel feedback system. The theory developed is formally limited to the area of linear transfer functions, but can be extended to a broader class of functions.

The principal equations of the theory developed are:

1. The relationship between oscillations U_r at test points Z_r and oscillations V_i at points of applications of force by sensors in system Y_i

$$\sum_{i=1}^{n} V_i \varkappa_{ir} = -(1-\alpha_r)\, u_r \tag{1}$$

(where χ_{ir} is the transfer function of the structure,
α_r is the required degree of attenuation),
allowing us to determine

$$V_i = \frac{\varkappa_{ir}{}^{(i)}}{\varkappa_{ir}} U_0,$$

Placing additional conditions on the frequency characteristics of the electromechanical channels of the system, we can determine the conditions of self-excitation of the multichannel feedback system and thus limit the maximum values of compensation of oscillations of the structure at the test points, which may not correspond with the fixed points. The primary quantities characterizing the structure (field) in the theory developed are the transfer functions K_{ij}; χ_{ir}, relating the values of oscillating parameters at points of excitation of oscillations Y_i by the radiators of the feedback system with the oscillating parameters V_i, V_r at points of placement of receivers x_j and correspondingly at the test points Z_r.

By representing the oscillating parameter of the primary field through U_j and U_r, the required degree of attenuation by α_r, we produce the relationship

$$\sum_{i=1}^{n} V_i \varkappa_{ir} = -(1-\alpha_r) U_r,$$

allowing us to determine the necessary parameter of compensation oscillations /86

$$V_i = -\frac{|\varkappa_{ir}|^{(i)}}{|\varkappa_{ir}|} U_0$$

Since

$$V_j = \sum_{i=1}^{n} V_i K_{ij} = \frac{U_0}{|\varkappa_{ir}|} \sum_{i=1}^{n} K_{ij} |\varkappa_{ir}|^{(i)},$$

then from the equation

$$V_i = \sum_{j=1}^{m} (U_j + V_j) \varphi_{ij},$$

relating the oscillating parameters V_i, created by the electromechanical feedback channels, to the summary oscillations at points of reception $U_j + U_j$ through the coefficients of conversion of the electromechanical feedback channels ϕ_{ij}, we can determine the values of ϕ_{ij} and further of the field created by the system outside the test points (V_p), expressing it through the values at the fixed test points

$$V_p(\omega_0) = \sum_{i=1}^{n} \frac{|\varkappa_{ir}|^{(i)}}{|\varkappa_{ir}|} \varkappa_{ip} U_0.$$

The conditions of self-excitation of a multichannel system and other necessary calculation parameters are correspondingly determined.

CERTAIN TYPES OF MULTICHANNEL SYSTEMS FOR COMPENSATION OF STRUCTURAL OSCILLATIONS

B. D. Tartakovskiy (Moscow)

The properties of multichannel systems used in certain cases when it is expedient to simplify a feedback system are studied. The compensated oscillations of the structure (sound field) are not made specific, which allows the generality of the results produced to be retained and allows them to be used when feedback systems are used for various specific purposes. Multichannel compensation systems of the following types are studied:

I. System consisting of identical receivers and sensors, symmetrically located in a homogeneous structure (field). By introducing certain average parameters of the system and structure (characterizing the set of channels of the system), the entire system can be represented by a certain equivalent feedback channel. /87

II. System with separate converters. In the general case, systems consisting of n radiators and m receivers, connected by mn independent electric channels characterized by conversion factors

$$\varphi_{ij};\quad i=1\div n;\quad j=1\div m\cdot$$

the number of converters is mn. It is demonstrated that when converters ϕ_{ij} are divided into two groups: ϕ_i and ϕ_j with the corresponding commutation, each of the receivers can be connected with each radiator without decreasing the generality while reducing the number of converters by a factor of $\frac{m \times n}{m + n}$.

III. System with electrically insulated channels. By eliminating electrical cross connection ($i \neq j$) in a system of general form $\phi_{ij}(t,j-n)$, we produce a system which consists of n single-channel subsystems (interconnected by oscillations of the structure (field)). The formulas for calculation of this system are significantly simplified since its possibilities for compensation of oscillations are comparatively little limited in comparison to a general system. If system III consists of identical links (system IV), the results of calculation become even more accessible for analysis.

Multichannel systems with a single receiver and many radiators, with many receivers and one radiator, as well as systems in which radiators and receivers

are combined, systems of sensors and control points, receivers and control points, as well as sensors, receivers and control points are also analyzed. The oscillation equations produced for these systems for the combination of structure plus feedback system are analyzed to determine the required parameters of the system to provide fixed compensation at test points and conditions for preventing self-excitation of the system. These problems are illustrated by calculation expressions for a two channel system (regardless of the type of oscillations of the concrete structure [field]). The results produced can be used for composition of an algorithm for control of a feedback system in a digital computer.

MEASUREMENT OF THE ATTENUATION FACTOR IN OSCILLATING SYSTEMS

A. I. Vyalyshev and B. D. Tartakovskiy (Moscow)

The method suggested allows the attenuation factor of systems with large /88 losses to be determined. It is based on determination of the shift in resonant frequencies in the mode of measurement of displacement, the oscillating velocity and acceleration. By using relationships for resonant frequencies of the oscillating system, the required formulas can be produced for determination of the attenuation factor.

The attenuation factor of an actual system may have an arbitrary frequency dependence, but individual sectors of the characteristic can be approximated by straight lines, corresponding to various types of loss in the system. Three models of frequency dependence of the dissipative force are used: R ~ d; R ~ v; R ~ a. Each model corresponds to its own formulas, determining the attenuation factors

Characteristic of Losses	$R \sim d$	$R \sim v$	$R \sim a$
$\eta =$	$\sqrt{\left(\frac{\omega_v}{\omega_d}\right)^4 - 1}$	$\sqrt{2}\sqrt{1 - \left(\frac{\omega_d}{\omega_r}\right)^2}$	$\sqrt{\left(\frac{\omega_v}{\omega_d}\right)^4 - 1}$
$\eta =$	$\sqrt{\left(\frac{\omega_a}{\omega_v}\right)^4 - 1}$	$\sqrt{2}\sqrt{1 - \left(\frac{\omega_v}{\omega_a}\right)^2}$	$\sqrt{\left(\frac{\omega_a}{\omega_v}\right)^4 - 1}$
$\eta =$	$\sqrt{\left(\frac{\omega_a}{\omega_d}\right)^2 - 1}$	$\sqrt{2}\sqrt{1 - \frac{\omega_d}{\omega_a}}$	$\sqrt{\left(\frac{\omega_a}{\omega_d}\right)^2 - 1}$

Using these formulas, we can also determine the form of losses in the system.

This method is convenient for measurement of the attenuation factor not only for concentrated but also for distributed systems. In the latter case, the problem is complicated by simultaneous excitation of a number of modes. In connection with this, corrections which depend on the form of the specimen and type of oscillations excited must be considered.

The loss factor of a damped rod with free ends was determined, in which bending oscillations were excited. The measurements were performed in three frequency ranges: 72-76 Hz, 126-132 Hz and 278-287 Hz. Using the values of resonant frequencies produced, the corresponding attenuation factors were calculated. For comparison, the attenuation factors were also calculated on the basis of the width of the resonant peak /89

Resonant area	η By width of peak	η By formula
72–76 Hz	0.19	0.17
125–132 Hz	0.22	0.2
278–287 Hz	0.26	0.21

As the losses increase, the divergence between these results increases as a result of the nonproportional increase in the width of the resonant peak, as a result of which the value of the attenuation factor determined by the ordinary approximate method is increased.

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STUDY OF DYNAMIC PULSE TYPE LOADS TRANSMITTED TO FOUNDATION BY NEW TYPES OF SHUTTLELESS LOOMS

Ya. I. Koritysskiy and R. I. Suchkova (Moscow)

In connection with the reequipping of textile enterprises with progressive equipment, in particular new types of looms, the necessity arises of developing methods of determining the peak dynamic loads transmitted by the machines to the foundation.

Together with theoretical methods, an experimental method is developed, /90 based on determination of loads using force measuring devices introduced as supports between the foundation and the base of the machine.

In order to record loads with minimum dynamic distortion, the machine-force meter system must satisfy the requirements placed on quasi-static instruments. In recording the loads from shuttle type looms with a four-link drive of the slay, the selection of force meter parameters represents no difficulties, since the rule of change of the perturbing function is periodic and can be represented as a sum of several harmonics.

This problem is more difficult to solve when force meters are used for new types of machines with cam-driven slay, the operation of which excites pulse type loads.

This report presents an operational method of evaluating the dynamic losses of instruments, in which due to incomplete damping the transfer function has significant oscillations. This method is particularly convenient for cases when the nature of the perturbing function is known only approximately in the form of an approximate graph. Recommendations are presented for selection of dynamic characteristics of the system of machine force meters and parameters of the meters.

DAMPING OF OSCILLATIONS OF RODS BY ELECTROMECHANICAL FEEDBACK

S. S. Korablev and V. I. Shapin (Ivanovo)

Regularities are found, explaining the operating principle of a controllable electromechanic oscillation damper operating over a broad frequency range.

Figure 1 shows one possible plan for the oscillation damper for a rod. The figure shows a rod of constant cross section F and length l.

The figure shows the control circuit, represented as a combination of elements L, R, C, an amplifier with gain factor S and actuating element AE, containing two electromagnets with a total clearance of Δ. (The inductance of the power winding of the AE is L_0, the resistance is R_0.)

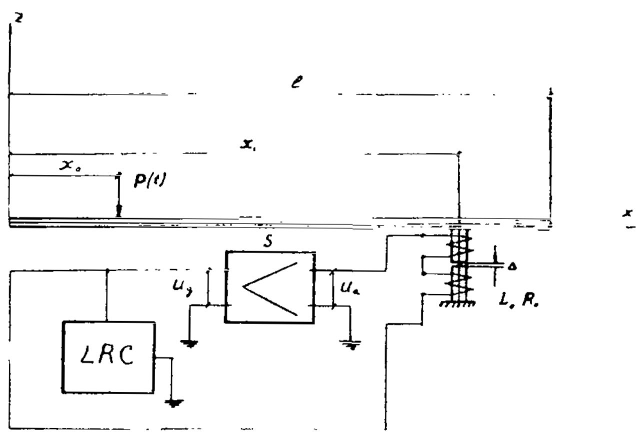


Figure 1.

On the assumption that within certain limits of tension and magnetic characteristics the AE can be looked upon as a linear quadrupole and the /91 operating mode of the amplifier and the AE is far from saturation, a mathematical model is constructed:

$$L_0 \frac{dy}{dt} + R_0 y + U \frac{\partial z(x_1, t)}{\partial t} = (s-1) L_1 \frac{dy}{dt} + (s-1) L \frac{d\psi}{dt} + (s-1) R_1 y$$

$$LC \frac{d^2\psi}{dt^2} + \psi = y - \frac{L}{R} \frac{d\psi}{dt}$$

$$EI \frac{\partial^4 z(x, t)}{\partial x^4} + \rho F \frac{\partial^2 z(x, t)}{\partial t^2} + \alpha \frac{\partial z(x, t)}{\partial t} =$$

$$= Uy\delta(x - x_1) + P(t)\, \delta(x - x_0)$$

Here $z(x,t)$ is a generalized coordinate, determining the vertical displacement of the rod, ψ is the current in the control circuit, y is the current in the anode circuit, $P(t)$ is the harmonic perturbing force, $\alpha \frac{\partial z(x,t)}{\partial t}$ is the resistance force to oscillating motion, U is the coefficient of electromechanical coupling, $\delta(x_0)$, $\delta(x_1)$ are the delta functions. The vibration-insulation conditions of the object (rod) are produced in the form:

$$Z_{el} + Z_x = 0 \tag{2}$$

$$U_j u_j(x_1)\vec{y} = -\vec{P}_j u_j(x_0) \tag{3}$$

Here the subscript j corresponds to the ordinal number of the natural form of oscillations of the rod; $\vec{y}, \vec{P}_j$ are the complex values of current in the anode circuit and the perturbing force, $u_j(x_1)$ and $u_j(x_0)$ are the values of the beam functions of the rod at $x = x_1$ and $x = x_0$, $Z_{el} = Z_x$ are the electrical impedances of the AE and control circuit. Thus, in order to decrease vibration it is necessary that the electrical impedances of the control circuit Z_{el} and converter Z_{el} be equal in magnitude and opposite in sine. /92

The complex expression of the amplitude of displacement of the rod is:

$$\vec{z}_j(t) = \vec{P}_j u_j(x_0)\left\{ m\omega_j^2\left[1 - \frac{\Omega^2}{\omega_j^2}\right] + j\Omega(\alpha + \alpha_{sup})\right\}^{-1}, \tag{4}$$

where

$$\alpha_{sup} = UU_j u_j^2(x_1)\left\{ R_0 + (1-S)\left(R_1 + \frac{\Omega^2}{Rx^2}\right) + j\Omega\left[L_0 + (1-s)\left(L_1 + \frac{1 - \frac{\Omega^2}{n^2}}{Lx^2}\right)\right]\right\}^{-\frac{1}{2}} \tag{5}$$

Here α_{sup} is the supplementary damping introduced to the system and determining, basically, the electrical parameters of the AE on the control circuit, ω_j is the frequency of natural oscillations of the rod, n is the partial frequency of the electrical circuit. As the number of the natural form of oscillations increases, the value of α_{sup} decreases. Consequently, the effectiveness of damping of oscillations of the rod is also decreased.

Figure 2 shows the calculated (continuous) and experimental (dotted lines) amplitude-frequency characteristics of the rod and notes the decreased level of vibrations. The frequency of the electrical circuit was tuned to the frequency of the perturbing force, while the AE was placed in the antinodes of the /93 corresponding forms of oscillation of the rod.

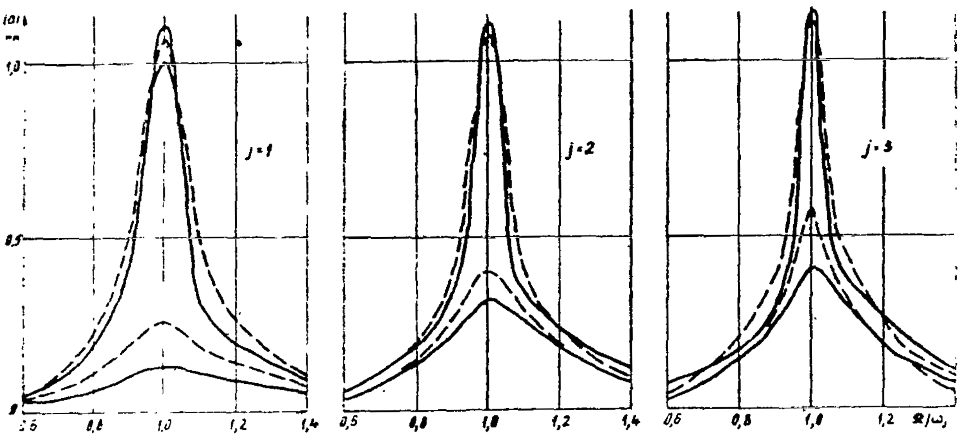


Figure 2.

STUDY OF NOISE IN VIBRATION AT INDUSTRIAL ENTERPRISES OF THE LITHUANIAN SSR

Yu. D. Valanchyaukas, M. E. Akelis and V. K. Naynis (Kaunas)

The study of production noise and vibrations at industrial enterprises of the Republic has shown that in designing new machines and equipment, in planning new production buildings and technological processes, the new methods of controlling noise and vibration are still not being sufficiently utilized.

At newly constructed enterprises, in many work areas the noise and vibration exceed the level permissible by the sanitary norms. The situation is similar in reconstructed enterprises in the Republic.

Investigations show that the noisiest enterprises in the Republic are the textile and construction materials enterprises, and that forging-pressing, metal working and shipbuilding shops have high noise level and high levels of vibrations.

At most enterprises, compressor devices are built into production rooms, and in some cases even into administrative rooms. Sound insulating materials are rarely used in the construction of air ducts.

The work of controlling the harmful effects of production noise and vibration at enterprises of the Republic should be performed in the following areas: First of all, decreasing noise and vibration at the source using technological, design and operational measures; secondly, decreasing the intensity of noise and vibration by sound and vibration absorbers, and thirdly, by using individual protection for workers.

DEVELOPMENT OF MULTIFUNCTIONAL VIBRATION MEASURING INSTRUMENT

Sh. M. Chabdarov, Ya. S. Uretskiy and V. V. Leont'yev (Kazan')

A multifunctional instrument has been developed and manufactured. The device allows a broad class of oscillation forms used in vibration testing to be modeled. /94

The device includes:

--a tunable harmonic generator;

--a polyharmonic generator;

--an oscillating-frequency generator with tunable middle frequency, band of oscillation and oscillating rate;

--a white noise generator;

--a vibrator amplitude-frequency characteristic equalizer;

--a wide band random vibration former with the required spectrum.

The report analyzes methods of forming wide band random vibrations with the required spectral characteristics.

A linear local-global method of forming random vibrations with the required spectrum is suggested, based on separate formation of the wide band portion of the spectrum and narrow band peaks and drops in the spectrum with subsequent addition of the signals formed.

Formation of the even portion of the spectrum is performed by a wide band amplifier with adjustable corrections in the low and high frequency areas, while formation of narrow band peaks and drops in the spectrum is performed by selective amplifiers, and the use of cophase-antiphase addition allows the number of selective amplifiers to be reduced and formation of peaks and drops to be performed by the same functional units.

THE PROBLEM OF PERFORMING STUDIES OF HARMONIC LOADS AND LOADS WHICH ATTENUATE WITH TIME

V. V. Karamyshkin (Moscow)

Under actual conditions, an object being tested is acted upon by a complex form of load or kinematic excitation, and it would be ideal to reproduce these excitations in experiments in order to place the test object under conditions /95 as close as possible to actual operating conditions. Due to technical difficulties, tests are generally limited to simple forms of dynamic loading--harmonic oscillations, free attenuation of oscillations (in particular from an impact).

It is assumed that a system being studied can be represented as a certain number of concentrated masses, connected with massless elastic elements. It is further assumed that experimental data are available on the amplitudes of oscillations at fixed frequencies at a number of points in the system being studied and that we must find at which points of the system harmonic effects must be applied to in order to reproduce the measured amplitudes. In the case of polyharmonic oscillations, based on the principle of superposition, the motion must be studied individually with respect to each frequency. The differential equations of motion of the system in the form solved relative to high order derivatives are written as

$$m_s \ddot{x}_s + \sum_{k=1}^{n} (b_{sk} \dot{x}_k + c_{sk} x_k) = Q_s \sin(\omega t + \varphi_s) \quad (s = 1,\ 2,\ \ldots,\ n), \tag{1}$$

where ω is the oscillating frequency, Q_s is the amplitude of the summary action on the s-th mass of the system, ϕ_s is the phase shift of the summary action. Replacing the right portion of (1) with $Q_s e^{i(\omega t + \phi_s)}$ and seeking the solution in the form

$$x_s = X_s e^{i\omega t} \tag{2}$$

we come to a system of algebraic equations for the complex oscillating amplitudes

$$\begin{gathered} p_{s1} X_1 + p_{s2} X_2 + \ldots + p_{s,\,s-1} X_{s-1} + (p_{ss} - m_s \omega^2) X_s + \\ + p_{s,\,s+1} X_{s+1} + \ldots + p_{sn} X_n = P_s \quad (s = 1,\ 2,\ \ldots,\ n), \end{gathered} \tag{3}$$

where we represent $p_{sk} = {}_{i}\omega b_{sk} + c_{sk}$, $k = 1, 2, \ldots, n$, $P_s = Q_s e^{i\phi_s}$. The desired solution will be the imaginary portion of expression (2).

Abstracting this, we can look upon system (3) as a certain linear transform of variables P_s to variables X_s. Solving the system (3) (which corresponds to an inverse transform), we produce the expression

$$X_s = \sum_{k=1}^{n} f_{sk} P_k \qquad (s = 1,\ 2,\ \ldots,\ n), \tag{4}$$

According to the theory of linear transforms, the values of X_s are determined unambiguously only if the number of external actions P_k is equal to the number of points at which the desired amplitudes are excited. Analyzing (3), we can produce conditions for determination of the minimum number of exciters. In /96
place of the quantities X_j, X_r, for example, we can analyze ωX_j, $\omega^2 X_r$, corresponding to reproduction of the desired velocity in acceleration of the j-th and r-th masses respectively. The structure of the equations is not changed.

Similar discussions are performed for beams with concentrated loads.

Free oscillations of a system with one degree of freedom are described by the expression

$$x = e^{-nt}(A \sin \omega t + B \cos \omega t). \tag{5}$$

The independent parameters here are n, A and B. Suppose the quantity ω is selected approximately as concerns the nature of the curve reproduced. Then due to the selection of n, A and B, we can produce the desirable displacement only at three selected moments in time. The motion caused by the pulses is added to the free attenuation.

Each intermediate instantaneous pulse adds two independent parameters--the intensity and application time of the pulse--and the possibility appears of assuring the desired displacement at two more moments in time.

NEW DEVELOPMENTS OF VIBRATION-MEASURING DEVICES WITH EXPANDED FREQUENCY RANGE

V. I. Petrovich (Moscow)

The frequency range of vibration-measuring apparatus with inertial vibration sensors is determined by the frequency characteristics of the seismic system of the vibration sensor. The dynamic range of such vibration sensors, the seismic system of which operates in the vibrometer mode, defined as the ratio of the frequency of highest resonances to natural frequency, does not exceed 20.

We can expand the dynamic range using positive feedback with respect to displacement (+OCC). The natural frequency of a vibration sensor with +OCC will be:

$$f_1 = \sqrt{f_0^2 - \alpha}, \quad (1)$$

where f_0 is the natural frequency of the seismic system without +OCC

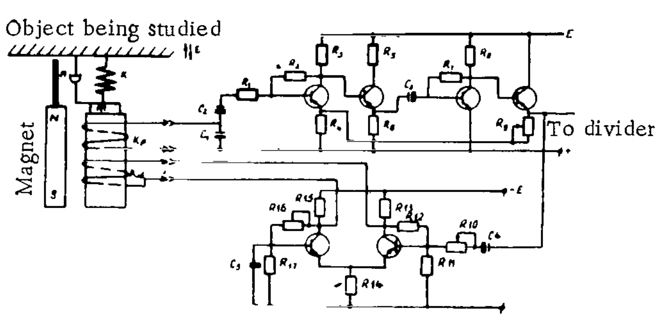
f_1 is the natural frequency of the seismic system with +OCC

α is the coefficient of depth of the +OCC,

then the dynamic range of the vibration sensor with +OCC is defined by the expression /97

$$K = \frac{f_b}{f_1} = \frac{f_b}{\sqrt{f_0^2 - \alpha}}, \quad (2)$$

from which we see that with increasing depth α, the dynamic range of the vibration sensor increases.



K = spring
П = damper
m = oscillating mass
k_p = working coil
$k_{o\delta}$ = feedback coil

Figure 1. Diagram of Vibration Sensor with +OCC.

A schematic diagram of a vibration sensor with +OCC is shown on Figure 1, the frequency characteristics of a vibration sensor with +OCC and without +OCC are shown on Figure 2.

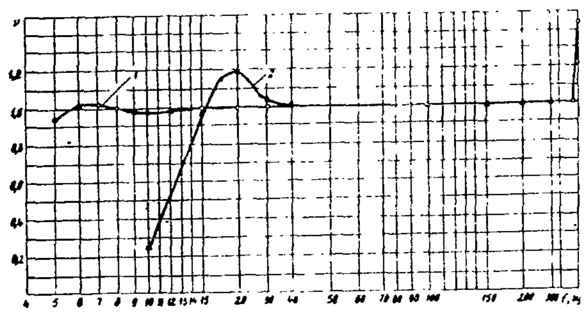


Figure 2. Amplitude-Frequency Characteristics of VD-6 Vibration Sensor. 1, With +OCC; 2, Without +OCC.

Using +OCC, the frequency range can be expanded by approximately five times.

Further expansion of the frequency range is limited by conditions of stability.

The new BIP-6 vibration-measuring apparatus has been developed, using vibration sensors with +OCC, operating in the 6-350 Hz frequency range in the 1-2000 μ amplitude range. /98

It should be noted that expansion of the frequency range using +OCC is not accompanied by a change in sensitivity of the sensor.

STUDY AND ELIMINATION OF INFLUENCE OF PRESSURE OSCILLATIONS IN PIPE SYSTEMS ON OPERATING PROCESS AND VIBRATION OF ELEMENTS OF PISTON COMPRESSOR STATION

V. A. Malyshev, B. M. Pisarevskiy and V. M. Pisarevskiy (Moscow)

1. In order to eliminate the harmful influence of pulsating gas flow, special pressure pulsation damping apparatus is included in pipelines.

However, its effectiveness is quite low, since the damping installations do not include devices for analysis of the influence of changes in the characteristics of the pulsating gas flow on the operating process and operational characteristics of the piston compressor station.

A method is suggested allowing the influence of pulsating gas flow on the operating process to be eliminated, consisting in matching the initial sector of the pipeline system, i.e., creating a load at the end of the initial sector so that only traveling waves can propagate in it.

The matching can be performed either by selecting the hydraulic resistance of the heat exchanger or oil and water separator elements of the first technological apparatus so that the specific (related to the area of the pipe) hydraulic resistance of the entire apparatus is equal to the wave impedance of the pipe, or by installing an additional element before the apparatus, the hydraulic resistance of which, when added to the hydraulic resistance of the apparatus, is equal to the wave impedance of the pipe. One necessary condition for possibility of matching is a predetermined volume of the technological apparatus.

Experimental investigations have indicated that the losses of power to overcoming the additional hydraulic resistance providing for matching of the sector are much less than the power consumed for compressing the gas when intensive /99
oscillations of pressure occur in the pipe. When the initial pressure is matched, the influence of oscillations of pressure in pipeline systems on the delivery of the installation is also eliminated, vibrations of the pipe systems are significantly reduced, loads on parts of compresser cylinders are decreased, i.e., the reliability of operation of the compressor apparatus is increased.

ANALYSIS OF HYDRODYNAMIC NOISE ARISING AS LIQUID FLOWS OVER ROUGH SURFACES

L. A. Sul'bi (Tartu)

As a liquid flows over a rough surface (course finishing, erosion, steps, etc.), turbulence and cavitation arise, which are frequently sources of intensive noise and vibration.

A cavitation generator (USSR Patent No. 2378217) was used to study hydraulic noise under various flow conditions.

The results produced in the study of hydraulic noise at flows of up to 50 m/sec over unevennesses of various shapes and sizes can be used as a basis for analysis of the operation of hydraulic structures. Comparison of the composition and intensity of the noise spectrum allows us to make judgments concerning the flow modes and condition of working surfaces in installations being tested.

METHOD AND INSTALLATION FOR STUDY OF PROCESSES OF VIBRATION MOVEMENTS OF PARTS AND ACCOMPANYING PHENOMENA

V. A. Povidaylo and R. I. Silin (L'vov)

In order to reveal the primary factors which have significant influence on the process of vibration movement of a part, a method was developed for experimental studies, allowing the wave picture of motion of a part in each cycle /100 to be determined, allowing the parameters of motion of the part, including phase angles of separation and encounter, the maximum separation of the part from the base and its phase angle, the presence and influence of attachment phenomena, the influence of elastic impact on the nature and picture of motion of the part, to be determined.

The total picture of motion of a part was produced by recording oscillograms of the vertical and longitudinal displacements of the part relative to the base on the tape of a magnetoelectric oscillograph, which required the development of a special installation and vibration measuring apparatus.

The experimental installation allowed simultaneous recording of the oscillations of the base, longitudinal displacement and motion in the perpendicular direction of the part relative to the base for several cycles. The installation is equipped with an automatic control system, allowing recordings to be performed during stable motion of the part, and contains a combination of high frequency inductive, induction and photoelectric sensors.

EXPERIMENTAL DETERMINATION OF CHANGES OF DYNAMIC CHARACTERISTICS OF HIGH-SPEED ROTORS OPERATING IN BALL-BEARING MOUNTS WITH PASSAGE OF TIME

V. I. Zdanavichyus and R. Yu. Vansevichyus (Kaunas)

The device and method developed allow the influence of the properties of a bearing on oscillations of the rotor in the radial and axial directions to be determined, the damping properties of a hydrodynamic oil film to be studied and phenomena of slipping of bearings in lightly loaded radial-thrust ball bearings to be studied at high shaft rotation rates. In order to study the motion of the rotor in ball bearings, conditions are created in the test stand near the natural conditions--a vacuum medium and high temperature.

In order to measure vibrations and displacements of the rotor, contactless capacitive differential motion sensors are used. Methods have been developed for contactless measurement of the resistance of a hydrodynamic oil film in high-speed ball bearings using circular capacitive current taps and transmission of current from the rotating shaft of the rotor to the bearing being tested using a special photoelement.

The method developed allows changes in dynamic properties of the ball /101
bearing as a result of weakening of the support of the bearing rings, lack of lubrication, etc., arising during rotation of the rotor, to be determined.

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CALCULATION OF VIBRATIONS OF DISTRIBUTED ELASTIC SYSTEMS BY FINITE ELEMENTS METHOD

V. P. Kandidov and L. P. Kim (Moscow)

1. In recent years, the method of finite elements has been developed for analysis of the vibrations of distributed systems and structures. In this method, the system is divided into a set of its parts, each of which is replaced by a certain model, called a finite element (FE). In contrast to the sector replaced, the FE has a finite number of degrees of freedom. The deformation and position of the FE in space are determined by the vector of generalized coordinates $\overline{q}(t)$, the components of which are the coordinates of the nodal points, which are common for neighboring elements. The conditions of their attachment are formulated at these points. As a result, the distributed system becomes a discrete model of FE.

2. In construction of FE, the displacement function w(r,t) of a sector of the initial system is limited by a class of certain basic functions $\overline{\psi}^{1}(r)$, as is done in the direct methods

$$w(r,\ t) = \overline{\psi}'\ (r)\ \|\ \alpha\ \|\ \overline{q}(t)$$

Using the dynamic principle of virtual displacements, the relationship is established between $\overline{q}(t)$ and the vector $\overline{Q}(t)$, the components of which are the generalized forces of interaction of the elements.

$$\|M\|\ddot{\overline{q}}(t) + \|K\|\overline{q}(t) = \overline{Q}(t) + \overline{F}(t)$$

F(t) is the vector of external forces, equal to $\overline{F}(t) = \int_R \|\alpha\| f(r,\ t)\ \overline{\psi}(r)\ dr^3$, where f(r,t) is the density of the external load.

$\|M\| = \int_R \|\alpha\|'\ \rho(r)\psi(r)\psi(r)\ \|\alpha\|\, dr^3$ is the matrix of inertia of the FE.

$\|K\|$ is the rigidity matrix. Its form is determined by the concrete expression for the energy of elastic deformation.

The set of these equations, written for each element, together with the conditions of their coupling and boundary conditions describe the dynamics of the model of FE.

3. As applicable to problems of twisting, bending and combined twisting--bending oscillations $\|M\|$ and $\|K\|$ matrices are produced for a number of

systems of basic functions. The change in elastic, inertial and geometric parameters along the length of the element is approximated by various analytic dependences.

In order to calculate the vibrations by the method of finite elements by digital computer, an algorithm is composed, allowing matrices of rigidity and inertia of the entire system as a whole to be produced. Then, known programs are used to find frequencies and forms of natural oscillations, or the reaction of the system to a predetermined action f(r,t).

4. As an example, the vibrations of a certain wing of an aircraft are analyzed. It is demonstrated that by improving the approximation of the elastic and inertial properties of a continuous sector by the corresponding element, we can increase the accuracy of calculation without increasing the number of degrees of freedom of the model of FE.

THE PROBLEM OF DECREASING VIBRATIONS OF ELECTRIC MACHINES

A. K. Bakshis, B. I. Shirvinskas and R. A. Dashevskiy (Kaunas)

Problems of dynamic balancing of rotors in a body in assembled form in the suspended state are studied. The conditions of equilibrium of an oscillating system (electric machine) are composed:

$$\left.\begin{aligned} M\frac{d^2 z_0}{dt^2}+cz_0 &= \sum P_i\cos(\omega t+\alpha_i) \\ I\frac{d^2\gamma}{dt^2}+k\gamma &= \sum M_i\cos(\omega t+\alpha_i) \end{aligned}\right\} \qquad (1)$$

where M is the mass of the oscillating system; /103

I is the moment of inertia of the oscillating system relative to the axis orthogonal to the plane of oscillations and passing through the center of mass;

Z_0 is the displacement of the center of mass of the system during oscillation;

γ is the angle of deflection of the axis of the rotor during oscillation;

c is the rigidity of the support in forward motion along the axis, located in the plane of oscillations and passing through the center of mass;

k is the rigidity of the support to twisting about the axis orthogonal to the plane of oscillations passing through the center of mass;

P_i is the centrifugal force of the rotating mass;

M_i is the moment of centrifugal force P_i, acting relative to the center of mass;

α_i is the angle of installation of the unbalanced masses;

ω is the angular velocity of the rotor;

t is time.

A method is presented for performing dynamic balancing of electrical machines in assembled form. The apparatus used is described and the basic data on the BP-1 balancing device are presented.

A block diagram of this device is shown on Figure 1.

The signal from the sensors, proportional to the imbalance, is sent to the decision device, which is designed to separate the planes of correction by an electrical method.

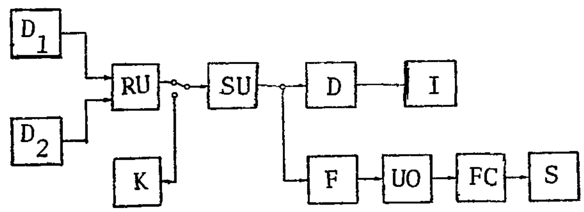

Figure 1. Block Diagram of BP-1 Balancing Device: D_1, D_2, Vibration sensors; RU, Decision device; K, Calibrator; SU, Selective amplifier; D, Detector; I, Arrow indicator; F, Phase reverser; UO, Amplifier-limiter; FC, Strobing pulse former; S, Strobe lamp.

The electrical signal is amplified by a narrow-band selective amplifier, rectified and sent to the needle indicator. The needle indicator shows the amplitude value of the imbalance signal.

The control of the strobe lamp is performed by the pulses formed from the /104
amplified and limited voltage of the imbalance signal. The phase inverter eliminates the inversion of phase introduced by the amplifier.

Calibration of the amplification of the selective amplifier is performed using the voltage of the calibrator.

Experimental results produced during balancing of electrical machines in assembled form are presented at the end of the work.

OSCILLATIONS OF A ROTOR RESULTING FROM INACCURACY OF MANUFACTURE OF BALL BEARINGS

A. P. Kul'vets (Kaunas)

In order to explain the action of a ball bearing on oscillations of a rotor, a dynamic model is studied, corresponding to a vertical rigid rotor, rotating in a prestressed radial-thrust ball bearing. The system has three degrees of freedom with three coordinates of the center of mass of the rotor x, y, z. The body and shaft are considered absolutely rigid, while the contacts of the rolling bodies with the external and internal rings of the bearing are considered nonlinear springs, the balls are considered massless. The oscillations of the rotor are excited by errors in the manufacture of the ball bearing. Each of the balls influences excitation of oscillations separately. The total influence can be greater or less due to the random nature of phases and amplitudes of the geometric errors in the shapes of each ball. Therefore, the "average" ball is analyzed, and its effect is a random function. Differential equations for the system are derived by Lagrange's method for the case when deformation of contacts resulting from the force of the preliminary interference increases the additional deformation resulting form displacement of the rotor and error in the form of rings and balls. The linearized differential equations for this case are solved. The studies performed allow us to determine the expected frequency spectrum of oscillations of the rotor and the frequency distribution of oscillation energies.

STUDY OF THE INFLUENCE OF RESONANT TWISTING OSCILLATIONS IN A MOTOR VEHICLE TRANSMISSION ON VIBRATION AND NOISE IN THE CAB

S. S. Stroyev and V. P. Belyayev (Chelyabinsk)

The influence of resonant twisting oscillations on the level of noise in a motor vehicle body was studied. The transmission of the Moskvich-412 automobile was replaced by an equivalent 7-mass dynamic system and the Holzer method was used to determine the frequencies of natural oscillations, which were compared with the frequencies of the exciting harmonic moments of various orders from the engine throughout its entire working speed range. This allowed analytic establishment of possible resonant oscillations of the motor and corresponding resonant speeds of the Moskvich-412 automobile, which were subsequently used to analyze the experimental data. In order to determine the zones of resonance excited by even harmonics of the model 412 engine, its vector and phase diagrams were studied. The most dangerous for the transmission are low order oscillations--single, double and triple-node. The results produced showed that the greatest oscillation of moment with a single-node form is observed on the crankshaft and half axle, with the double-node formed--on the wheel and half axle and with the triple-node formed on the primary shaft of the transmission. Furthermore, calculations established that under actual operating conditions, resonance may develop in second gear at 36.2 km/hr, 47.9 km/hr and 72.5 km/hr, and in third gear at 50.6 km/hr and 67.5 km/hr. The calculated values of frequencies are compared with the vibration spectrum. The experimental confirmation of these results was performed on a test stand with rotating loaded rollers using strain-gages. /105

Vibrations in the body of the vehicle during the experiment were recorded by a specially designed device allowing the oscillations of the body over the front and rear wheels to be measured.

APPLICATION OF DIMENSIONAL ANALYSIS TO STUDY OF VIBRATION ACTIVITY OF PISTON ENGINES

V. L. Kalishevskiy and A. M. Opolchentsev (Moscow)

The study of vibration activity of mechanisms in analytic form is a complex task and cannot be performed completely without significant simplifications. In an experimental study, the partial dependences of individual parameters are /106 produced. It is difficult to compare the results of tests of mechanisms of various types.

Analysis of the conditions of similarity of mechanisms as to vibration activity allows us to estimate the influence of various technological, design and mode parameters of the machine on its vibration activity, to determine the volume and method of experimental study.

For a system described by the differential equation

$$M\ddot{q} + R\dot{q} + Cq = P_a \sin \omega t \tag{1}$$

we can compose the following dimensionless numbers:

$$K_1 = P_a/M\ddot{q}_a; \quad K_2 = C/M\omega^2 = \omega_0^2/\omega^2; \quad K_3 = C/R\omega. \tag{2}$$

The criteria K_2 and K_3 characterize the properties of the system (inertial, rigid and damping characteristics) and determine the similarity of the data of the system. Complex K_1 expresses the relationship between the perturbing force and the vibration which it causes.

Since the level of vibration is measured relative to a threshold vibration acceleration q_0, we can write:

$$L = f(\ddot{q}_a/\ddot{q}_o) = f(P_a/M\ddot{q}_o; \ \omega_0/\omega; \ C/R\omega). \tag{3}$$

Consequently, for this system the dimensionless number for vibration activity is defined as

$$L = idem$$

where

$$\frac{P_a}{M\ddot{q}_0} = idem; \qquad \frac{\omega_0}{\omega} = idem; \qquad \frac{C}{R\omega} = idem. \tag{4}$$

In the general case for a complex system

$$L=f\left(\frac{P_{a1}}{M_1\ddot{q}_0};\ \frac{P_{a2}}{M_1\ddot{q}_0};\ldots\frac{P_{a1}}{M_2 q_0};\ \frac{P_{a2}}{M_2\ddot{q}_0};\ldots\frac{P_{am}}{M_j\ddot{q}_0};\ \frac{\omega_{01}}{\omega_1};\right.$$
$$\left.\frac{\omega_{02}}{\omega_1};\ldots\frac{\omega_{01}}{\omega_2};\ \frac{\omega_{02}}{\omega_2};\ \ldots\frac{\omega_{0l}}{\omega_k};\ \frac{c_n}{R_e\,\omega_k};\ \frac{\omega_k}{\omega^{(1)''}}\right), \tag{5}$$

where the relationship ω_k/w'' considers ω_k = var and $\omega^{(1)''}$ = 2π is the unit frequency introduced for retention of nondimensionality. The condition of unambiguity must be fulfilled, including analogy of the designs of the mechanisms, technology of manufacture, test conditions, operating conditions, etc.

For the piston pumps studied, considering $L = f(P_{am}/Mq_0)$, this function /107 expresses the empirical dependence of the overall levels of vibration of pumps on the vibration activity factor

$$L=72+14\lg\Phi_B=72+14\lg\frac{P_H Q}{M}, \tag{6}$$

where the product of delivery pressure P_H times delivery Q is used as a characteristic of the perturbing force.

Analysis of dependence (5) is simplified if we study the similarity conditions with respect to individual main vibration sources. In the general case

$$L=f\left(\frac{P_a f^2}{G f_0^2};\ \frac{P_a f}{G f_0};\ \frac{f}{f''}\right) \tag{7}$$

For certain known sources of vibration in piston pumps, the following dependence can be written:

$$L=f\left(\frac{\alpha G_H r f^4}{G\ddot{q}_0 f_0^3};\ \frac{\alpha G_H r f^3}{G\ddot{q}_0 f_0};\ \frac{mv_k^2}{f_k ER};\ \frac{\beta p_H F f^2}{G f_0^2};\ \frac{\beta p_H F f}{G f_0};\ \frac{f}{f^{(1)}}\right), \tag{8}$$

which is simplified for the pumps being studied with f_0 = const:

$$L=f\left(\frac{\alpha G_H r f^4}{G\ddot{q}_0 f^{(1)3}};\ \frac{m r_k^2}{f_k ER};\ \frac{\beta p_H F f}{G f^{(1)}};\ \frac{f}{f^{(1)}}\right) \tag{9}$$

As a result of tests, the following empirical dependences have been produced:

$$L=66+14\lg K_{PS}=66+14\lg\frac{\beta p_H F f}{G f^{(1)}} \tag{10}$$

$$L = 20 \lg K_{k l} \cdot 10^{12} = 20 \lg \frac{m v_k^2}{f_k E R} \cdot 10^{12} \quad (11)$$

Analysis of the similarity conditions of piston pumps as to vibration activity and empirical dependences of levels of vibration on the complexes Φ_B, K_{PS} and K_{kl} were used to study means of creating low-noise pumps and to evaluate their vibration activity during the planning stage.

STUDY OF VIBRATIONS IN THE DIRECTION OF INCREASING AND DECREASING TIME

V. L. Ragul'skene (Kaunas)

In order to calculate the behavior of oscillating systems in the direction of increasing and decreasing time, it is suggested that differential equations /108 be used, allowing a significant simplification of the definition of free oscillations, areas of capture, areas of existence, etc.

If the motion of a system in the direction of increasing t is described by the following differential equation

$$F\left(\frac{d^n x}{dt^n}, \frac{d^{n-1} x}{dt^{n-1}}, \ldots, \frac{dx}{dt}, x, t\right)=0, \quad (1)$$

then the motion in the direction of decreasing time is described by the following equation

$$F\left[(-1)^n \frac{d^n x}{d\vec{t}^n}, (-1)^{n-1} \frac{d^{n-1} x}{d\vec{t}^{n-1}}, \ldots, \frac{-dx}{d\vec{t}}, -t\right]=0, \quad (2)$$

where x is a generalized coordinate, t is time in the increasing direction, $\vec{t}$ = -t is time in the decreasing direction.

Thus, the motion of a single-mass vibration impact system in the direction of increasing time is described by the following equations

$$\frac{d^2 x}{dt^2}+2h\frac{dx}{dt}+h_0 \operatorname{sign}\frac{dx}{dt}+p^2 x=A\sin\omega t,$$

$$x>x_y,$$

$$\left(\frac{dx}{dt}\right)^+=-R\left(\frac{dx}{dt}\right)^-, \qquad x=x_y,$$

and in the direction of decreasing time

$$\frac{d^2 x}{d\vec{t}^2}-2h\frac{dx}{d\vec{t}}-h_0 \operatorname{sign}\frac{dx}{d\vec{t}}+p^2 x=-A\sin\omega t,$$

$$x>x_y,$$

$$\left(\frac{dx}{d\vec{t}}\right)^+=-R\left(\frac{dx}{dt}\right)^-, \qquad x=x_y,$$

where the "+" and "-" in the first derivatives represent the velocities at the moment before impact and after impact.

The solution of the equations in the directions of increasing and decreasing time by computer allows universal investigation of the properties of the dynamics of a number of vibration impact and nonlinear systems.

CONTACTLESS METHOD OF MEASURING VELOCITY OF IMPACTING ELEMENTS

Yu. M. Vasil'yev, A. V. Mikhailov, Yu. K. Skvortsov, V. A. Smirnov (Moscow)

The essence of the method being studied is that the cavity in a body in /109
which the impacting element moves is excited at the corresponding wave length λ_0 as a high-frequency cavity resonator, retuned periodically as the impacting element moves due to the change in longitudinal size of the resonator, and the frequency of retuning of the resonator is recorded. The resonance occurs for a cylindrical cavity resonator at moments when the length of the resonator is equal to a whole number of half waves $\lambda_b/2$, where:

$$\lambda_b = \frac{\lambda_0}{\sqrt{1-\left(\frac{\lambda_0}{\lambda_{cr}}\right)^2}}$$

λ_b is the wave guide number of the wave in the cylinder, which is analyzed as a circular wave guide;

λ_{cr} is the critical wave length for the type of wave selected.

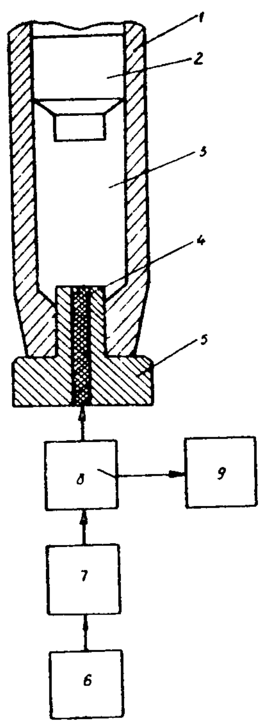

Figure 1. Block Diagram of Measuring Installation.

The frequency of retuning of the cylindrical cavity resonator F (1/time between neighboring resonant peaks) depends on λ_b and the rate of motion of the impacting element V:

$$F = \frac{V}{2\lambda_b}$$

Recording frequency F allows us to find the distribution of velocity on the path of the impacting element.

Figure 1 shows the diagram of measurement of the velocity of the impacting element in an air hammer. Impacting element 2 moves in body 1, changing the length of the cavity resonator 3, which is excited at wave length λ_0 = 8 mm as a cylindrical cavity resonator by circular wave guide 4, passing through the insert 5 and fed by generator 6 through attenuator 7 and guiding tap 8. Inset 5 is inserted into the body in place of the blade of the air hammer. In order to retain the operating mode of the tool,, the circular wave guide 4 is filled with a radio-transparent dielectric (e.g., teflon).

In case of a complex dependence of adjustment frequency on speed of the impacting element, static calibration must be performed, i.e., the dependence of the signal from the detector on the position of the impactor in the cylinder is measured and used in processing the results of measurement. /110

STUDY OF MECHANICAL OSCILLATING SYSTEMS USING NATURAL VIBRATORS

B. T. Sheftel' and Yu. A. Kamendrovskiy (Saratov)

The oscillating system of a ball bearing with a radial clearance (Figure 1) in which the kinematic exciting factor consists of errors in the form of the rolling tracks, so-called waviness, was studied. The waviness was expanded /111 by electronic apparatus in a Fourier series, and one of the harmonics of the waviness was used as a vibrator. Due to compact elasticity between balls and rings and bending oscillations of the outer ring, the vibration acceleration at frequency f_{ki} will be

$$W_{ki\mathrm{II}} = B(f)\, W'_{ki\mathrm{I}},$$

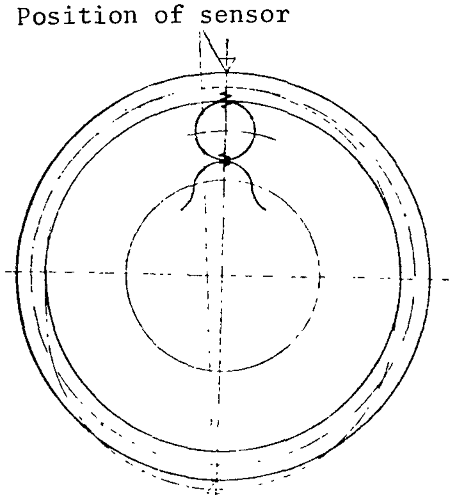

Figure 1.

where

$$B(f) = \frac{W_{ki\mathrm{II}}}{W_{ki\mathrm{I}}} -$$

is the coefficient of dynamism.

By changing the rotating speed of the bearing, with a high value of i we can produce a rather broad range of change of vibration frequency. By measuring acceleration W_{ki} of the ring at various frequencies, it is easy to calculate the values of the coefficient of dynamism and construct on the basis of these values the amplitude-frequency characteristics of the system.

The experiments were performed on a test stand. The inner ring of the bearing was seated on a spindle, rotated at a regular speed, while the non-moving outer ring was loaded in the radical direction. In order to measure the vibrations, an induction velocimeter sensor was used, the body of which was /112 rigidly fastened to the base of the test stand, while the probe rested on the top of the outer ring of the ball bearing (in the loaded area) if the frequency of free oscillations of the system as a result of contact elasticity of the part was being determined or on the bottom if the frequency of free bending oscillations of the external ring was being determined.

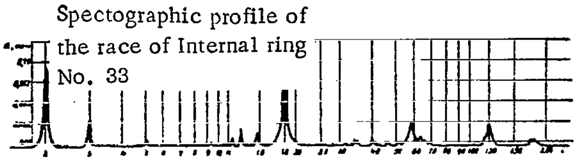

Figure 2.

The objects of investigation were No. 306 ball bearings. The vibrator was the 120-th harmonic of waviness of the race of the internal ring No. 33, the amplitude of which was 0.05 μ (Figure 2). Various bearings were assembled with this same inner ring and their amplitude-frequency characteristics were measured by the method described.

THE EXCITATION OF PERIODIC AND RANDOM TORSIONAL OSCILLATIONS IN ROTATING SYSTEMS

R. Yu. Bansevichyus and K. M. Ragul'skis (Kaunas)

The device used for application of loading torque of any sign and form to a rotating system consists of a geared rotor and stator with a multiphase winding, fed by pulses of predetermined duty factor by a special commutator controlled by a photoelectric sensor which locates the angular position of the rotating system. The load of any type is formed of sectors of the static characteristics of the stator-rotor system during rotation of the rotor of the excited system. This allows torsional oscillations of the rotating system to be excited in a frequency range beginning at 0 Hz, the frequency and amplitude of the exciting oscillations being independent of the angular speed of rotation of the excited system and controlled by changing frequency and amplitude of the supply voltage applied to the commutator.

Random oscillations with fixed spectral density are excited by attachment /113 of a commutator through the amplifier to a random function generator.

The upper boundary of the frequency range is determined, depending on the structural parameters of the device and the angular rotating speed of the excited system. The dependence of the maximum value of applied torque on the moment of inertia of the attached system is determined. The results of experimental studies are presented.

IDENTIFICATION OF CHARACTERISTICS OF DYNAMIC SYSTEM OF CYLINDRICAL GRINDER BY "BLACK BOX" METHOD DURING GRINDINGS

Yu.-V. P. Astrauskas and R. A. Ionushas (Vil'nus)

The dynamic characteristics of the "grinding process" and multiple-bushing hydrodynamic friction bearing of the spindle of a grinding wheel with respect to input and output signals applied during the grinding process are studied.

In order to determine the transfer function of the "grinding process" link as a "black box" the input and output signals are studied, the interrelationship between them is determined, as well as the presence of feedback and noise and their influence on the input and output signals of the link.

A simple expression is produced for the complex transfer function of the "grinding process" link as a diagnostic object, relating the spectral density of input and mutual spectral density of input and output signals.

In determining the transfer function of the hydrodynamic friction bearing on the basis of the functional diagram of the bearing, the nature of couplings between spindle and grinding head was studied. Elastic and damping couplings on each bearing bushing were used. In order to analyze the dynamic characteristics of the friction bearing, it was preliminarily treated as an oscillating link with damping, with two inputs and two outputs.

The input signals are the irregularity of the geometry of the neck of the grinding spindle in circular cross section and the oscillations of the grinding forces, while the output signals are the oscillations of the axis of rotation of the spindle in two perpendicular directions.

The force oscillations from the electric motor of the drive of the grinding /114 wheel are taken as noise acting on the input of the link, while the oscillations from the hydraulics of the machine are taken as noise acting on the transfer link.

The statistic chacteristics of the input and output signals determined are used to produce a transfer function (dynamic characteristic) of the hydrodynamic friction bearing.

SPECTRAL-CORRELATION ANALYSIS OF VIBRATIONS OF AVIATION ENGINE UNDER TEST STAND CONDITIONS

S. G. Gershman, N. G. Dubravskiy, V. I. Povarkov (Moscow)

Vibrations of the body of a jet aircraft engine type RG-9B were studied. The signals from four vibration sensors installed at various points on the body of the engine were amplified by a wide band amplifier and simultaneously recorded by a five channel magnetic recorder. These recordings were then reproduced for subsequent analysis.

Spectral analysis was performed by a spectral analyzer with a resolving capacity of 6 Hz in the 20-20,000 Hz frequency range. Correlation analysis was performed by an analog correlator with continuous change of delay time. Recordings of vibration of the engine in four states were produced and analyzed:

1. Full operating conditions;
2. Damaged--with one turbine blade cut off at one-third of its length;
3. Damaged--with a turbine blade cut off to one-sixth of its length;
4. Damaged--with a defect on the race of the middle bearing.

The power spectra of vibration of the engine in proper condition consists of continuous, smooth, wide band noise with superimposed discrete tones, related to the rotating frequency of the rotor. Some of these tones represent the noise of rotation of compressor or turbine stages. Some discrete tones are combination frequencies of noises of rotation of various stages.

The similarity of the dependences of spectral vibration densities for sensors remote from each other is less than the similarity of spectra from neighboring vibration sensors.

The analysis of vibrations for various operating cycles of the engine /115
showed good reproducibility of spectrograms. The defects intentionally introduced to the engine caused certain changes in the spectrograms and correlograms.

For example, in the experiment when one blade in the second stage of the turbine was cut off to one third its height, a change was noted in the low frequency area of the wide band noise, with an increase in the harmonic at the rotating frequency of the rotor, plus the appearance of side bands around the noise frequency of rotation of the second stage of the turbine. These changes also influence the autocorrelation analysis.

In the experiment with a turbine blade cut off to one-sixth, similar changes were observed, but expressed more weakly. They can be seen precisely only by correlation methods in combination with preliminary filtering of the signal in the selected frequency band. In order to allow correlation evaluation of changes in the parameters of the spectrograms and correlograms, a certain function is introduced to the work, a measure of the change which characterizes the degree of damage.

In the experiment with a defective bearing, a change was noted in the distribution of spectral density of the continuous noise which is obviously a result of the disassembly of the engine which was required to install the defective bearing. Various harmonics of the rotor rotation frequency increased. Spikes appeared on the spectrograms at frequencies which were multiples of the cycling frequency of balls in the external bearing circle. In most cases, they were small in amplitude and therefore, a certain statistical analysis was performed, demonstrating the nonrandom origin of these noise bursts.

The harmonics of the cycling frequency of balls in the external ring of the bearing were also revealed by the correlation method. However, the use of the correlation method was hindered by the presence of stronger secondary discrete bursts, the frequency of which differed little from the frequency of cycling of the balls.

REGRESSION ANALYSIS OF NOISE OF AN AIRCRAFT ENGINE

S. G. Gershman, V. D. Svet, V. I. Povarkov (Moscow)

Experimental results are presented from the measurement of the regression lines, coefficients of correlation relationships and mutual correlation coefficients for the vibration noises of an RD-9B aircraft engine. Recordings of vibrations of the engine in two states were analyzed: fully correct operating /116 state and defective state (one turbine blade cut off to one-third height).

The correlations between discrete components, as well as relationships between envelopes in various areas of the high frequency spectrum were studied. The selection of spectral sectors was performed on the basis of spectrograms made earlier. As a result of these measurements, it was established:

1. In the defective state of the engine, there are considerably nonlinear correlations between the low frequency components of the spectrum. In the normal engine, no nonlinear correlations were detected.

2. There are also nonlinear and linear correlations between the low frequency components and the envelopes of the high frequency areas of the spectrum. In the normal engine, no nonlinear correlations were noted.

3. Quantitative processing and the numerical results of the coefficients of the regression lines, coefficients of correlation ratios and mutual correlation showed that the relations between various components of the noise spectrum of the aviation engine are close to functional. The strongest relationship appears between the main rotating frequency of the rotor and its second and third components (with defective engine operation), allowing us to limit ourselves in analysis to the first four terms of the series of the polynomial regression.

The determination of the regression lines and coefficients of correlation ratios and mutual correlation was performed using methods and apparatus analyzed in [1].

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1. Gershman, S. G., V. P. Prikhod'kol and V. D. Svet, "Some Problems in the Application of the Two-Dimensional Probability Distribution Function for Analysis of Acoustical Noise and Signals" (see this collection).

SOME PROBLEMS IN THE APPLICATION OF THE TWO-DIMENSIONAL PROBABILITY DISTRIBUTION FUNCTION FOR ANALYSIS OF ACOUSTICAL NOISE AND SIGNALS

S. G. Gershman, V. P. Prikhod'kol, V. D. Svet (Moscow)

In many problems of investigation in acoustical noise, it is frequently sufficient to know the two-dimensional distribution function $W(x,y,\tau)$ of processes $x(t)$ and $y(t)$ to construct a statistical model of the phenomenon being studied. The expediency of determining $W(x,y,\tau)$ arises for a number of reasons.

Actual acoustical noises frequently do not follow the normal distribution, /117 which indicates the specifics of their noise formation, while the parameters characterizing the deviations of these distributions from normal contain useful information. Another specific feature of the two-dimensional distribution function is that it contains all necessary parameters for estimation (in the general case) of the arbitrary stochastic relationship between two random processes.

In particular, determination of $W(x,y,\tau)$ allows us to use methods from the general theory of correlation, including regression and dispersion analysis. Furthermore, for many problems measurement of $W(x,y,\tau)$ allows us to produce a large number of statistical parameters, necessary for description of the phenomena being studied. Therefore, the measurement of the joint distribution function $W(x,y,\tau)$ is of definite interest in problems investigating the physical nature and mechanism of noise formation, machine diagnosis, the study of nonlinear objects, etc.

A practical method of measuring $W(x,y,\tau)$ and its parameters is analyzed; a functional diagram of a specialized device is presented. In combination with digital computers, the device allows the following measurement problems to be solved: a) preliminary selection and classification of processes being studied by accumulation and visualization of $W(x,y,\tau)$ in real time; b) input of a large mass of information to a digital computer; c) determination of the required processing algorithm, thereby eliminating the basic difficulty in application of modern digital computers for massive statistical processing.

Experimental results are presented for measurement of the two-dimensional distribution function and their parameters for various acoustical signals. The results produced confirm the effectiveness of using $W(x,y,\tau)$ in problems involving analysis of acoustical noise and signals and the feasibility of this method of measurement.

ONE CONTACTLESS METHOD OF STUDYING THE NATURAL OSCILLATIONS OF ELASTIC STRUCTURES

B. D. Tartakovskiy, V. P. Shmal'chenko, M. M. Efrussi (Moscow)

During determination of the natural frequencies and losses of elastic oscillations of solid structures, errors arise due to the attachment of contact electromechanical converters to the structures. This defect is not true of the method of excitation of natural oscillations of resonant structures by a radiator located in the surrounding medium, and measurement of the frequency characteristics of the structure and the rate of attenuation of natural oscillations by a sensor also located in the surrounding medium. /118

Keeping in mind the more favorable ratio of impedances of elastic structures (plates, rods, envelopes and their combinations) with the wave impedance of fluids in comparison with gases, we developed a method for measurement of elastic structures submerged in a liquid (water). The specimens studied were thin aluminum spheres and cylinders. The sound field was created by a radiator located at a distance of 0.5 m from the specimen, the sound pressure receiver was placed at a distance of 1 m from the specimen. With a continuous change in the frequency of excitation, the resonant frequencies of oscillation of the specimens were determined, then the oscillations of the radiator were cut off at these frequencies, thus recording the process of attenuation at each of the resonant frequencies using a strip chart recorder. Figures 1 and 2 show examples of the results produced.

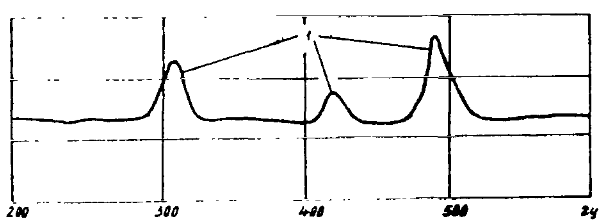

Figure 1. Frequency Characteristic of Oscillations of a Specimen: 1, Resonances of specimen.

On Figure 1 we show a sector of the frequency characteristic of oscillations of a specimen. Excitation of the attenuating oscillations of the specimen was performed at the resonant frequencies of the specimen thus determined. The recording of the process of attenuation of natural oscillations of the specimen being studied on the strip chart recorder is shown on Figure 2. Observing this process from left to right, we can see at first the summary sound field of the radiator and specimen. After the radiator is switched off, since its attenuation occurs over a significantly briefer time than attenuation of the specimen, the damping oscillations of the specimen remain, decreasing to /119 the level of noise. This method allows information to be determined on such mechanical characteristics of objects being studied as the mechanical loss factor, resonant characteristics, etc. without any noise related to contact excitation and measurement of vibration of the specimens.

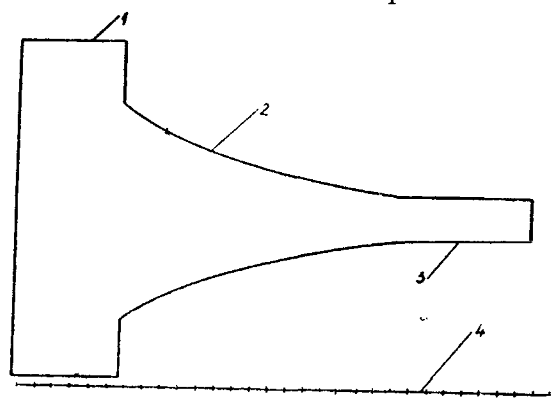

Figure 2. Recording of Process of Attenuation of Specimen: 1, Summary signal of radiator and specimen; 2, Attenuation of specimen; 3, Noise; 4, Time marks.

DESIGN OF VIBRATION-PROTECTIVE SYSTEMS WITH RANDOM VIBRATION ACTION

P. N. Ilgakois (Kaunas)

In this work, vibration-protective systems are calculated with random oscillations applied to the base with one predominant oscillating frequency.

During planning of vibration-protective systems for precision equipment, electronic apparatus, it must be considered that the vibration effects are not always deterministic; therefore, they cannot be described by deterministic functions of time and the vibration effects must be looked upon as a random process.

The problem of calculation of the optimal values of the natural frequency of a vibration-isolated object and the amount of damping when vibration-isolating supports with a specific elasticity curve are used are analyzed with /120 random oscillations of the type mentioned above as the excitation.

The method of calculation presented allows the expediency and effectiveness of the use of a vibration-protective system to be estimated.

CERTAIN PROBLEMS OF MACHINE ACOUSTICS

M. D. Genkin and V. I. Sergeyev (Moscow)

The basic positions of machine acoustics, as one new area in the general theory of machines, have been presented in preceding works.

At the present time, it seems expedient in the solution of applied problems to perform investigations in the following main areas.

First of all, methods must be developed to reduce the vibration-acoustical activity of the machines, which must be based on study of vibration-acoustical parameters of the most representative objects of various types of machine building, investigation of their vibration-acoustical fields, consisting of interconnected units for excitation of oscillating energy and sectors of structures (bodies, supports, etc.), which, reacting to this excitation, are a unique "vibration-acoustical load for the exciting units." It should also be kept in mind that these structural sectors are in turn secondary sources of excitation for attached parts, thus creating a system of interconnected vibration guides--channels through which oscillating energy propagates, working with external loads.

Secondly, the methods of solution of this range of problems must be based on studies of the physical principles of vibration acoustical excitation in machines and their elements; investigations in this connection of the influence of the random nature of loading with statistical distribution of the values of parameters of systems, investigation of the deformation of links in kinematic circuits, study of the oscillations of complex systems of mechanisms and structures resulting from this excitation, identification of these oscillating states, development of theories for modeling the dynamic and acoustical processes, as well as methods for synthesis and optimization of machine structures on the basis of the criterion of minimum vibration acoustical activity, study of the processes of dynamic interaction of mechanisms and structures, investigation of the propagation and absorption of wave energy in discrete and continuous heterogeneous structures, study of the dispersion properties and problems of diffraction in mechanical structures.

One effective means for controlling vibrations is a change in the design /121 of elements of machine units, including supporting structures.

Another promising method for reducing the level of vibration acoustical activity of machine units as a whole (by shutting off the flow of oscillating energy within each unit) is the use of active antivibrators, creating counter-phase vibration. Particularly great effects can be achieved by combining active antivibrators with elastic elements, i.e., passive shock absorbing systems. This provides simultaneously rather rigid fastening of mechanisms and a high degree of vibration insulation, as characteristic for soft suspensions.

As the development of general methods has shown, a decrease in the vibration acoustical activity of machine units should be based on study of the properties of the vibration acoustic fields of the units. In this connection, there is great significance in the development of experimental methods and apparatus for investigation of the corresponding characteristics of the vibration acoustical fields.

Usually in complex mechanical systems, the transmission elements perform oscillations in various forms and therefore one of the most important problems is the division of special oscillations of rotating parts into components.

A no less important task is the measurement of the band parameters of multi-dimensional oscillating mechanical systems in order to investigate the dynamic interaction of mechanisms and foundations, and also to determine the levels of oscillating energy transmitted by a mechanism to the foundation, i.e., determine the degree of vibration acoustical activity of the machines. This problem is directly related to the solution of a combination of methodological problems directed toward creation of new special measurement apparatus and estimation of its accuracy.

The vibration acoustical properties of machines are also used for diagnostic purposes.

As we know, an idea of the reliability and efficiency of a machine, as well as a prediction of its future behavior can be produced by observation of its operating condition. The number of parameters which determine the state of a machine, generally speaking, is infinite. However, we can separate among these the principal parameters, which determine the quality of the working process. Direct measurement of these parameters is generally too difficult and expensive, and sometimes is simply impossibie. Therefore, attempts are made to measure them indirectly.

Of all indirect indicators which can be used to judge the operating condition of the machine (temperature, various mechanical indicators, e.g., oil pressure, radiation, sound, etc.) the most effective is the sound signal. /122

There are many systems for acoustical dianosis, adapted for various types of machines. They all use the physical specifics of a definite type of machine and are unsuitable for others. Externally, all these systems differ strongly from each other. But the number of characteristics used is limited.

Acoustical diagnosis is one of the most important methods of investigation in the area of machine acoustics.

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DETERMINATION OF THE PARAMETERS OF MECHANICAL OSCILLATING SYSTEMS ON THE BASIS OF THE AMPLITUDE-FREQUENCY CHARACTERISTICS AS A MEANS OF VIBRATION DIAGNOSIS OF MACHINES

M. D. Genkin, A. A. Zhirnov, et al. (Moscow)

Gear transmissions are widely used in many branches of machine building, and the quality of their manufacture and assembly, as a rule, determines the vibration and noise characteristics of extremely complex and important units. The results of detailed testing of the quality of manufacture of individual gears or test stand experiments of individual units as assembled frequently do not correspond to their operating conditions when they are actually in use. A method is suggested for determining the actual values of error in gears during operation, allowing estimation not only of the magnitude of tooth and cyclical errors, but also of the quality indicators such as, for example, deviation in the size of the contact spot of a tooth pair.

The method is based on comparison of the experimentally produced frequency spectrum with the calculated vibration spectrum of a dynamic model. The dependence is thus established between the components of the frequency spectrum and the elements of the dynamic model reflecting specific parts of the structure.

In particular, the component frequencies of the spectrum determined by /123
rigidity of each of the meshed gears in the unit being studied can be defined.

Results are presented from determination of the actual gear and cyclical errors in various operating modes of transmissions.

PROBLEMS OF IDENTIFICATION OF PARAMETERS OF DYNAMIC SYSTEMS BASED ON TAPE DRIVES

A. Navitskas, K. Ragul'skis, O. -M. Skurkayte (Kaunas)

This work presents one method for determination of the optimal mechanical and dynamic parameters in studies of complex dynamic systems containing both electronic and mechanical parameters. Such systems include various types of magnetic recording apparatus, an evaluation of the accuracy of information transmission (electronic parameters) of which is closely related to the function of the tape drive mechanisms and other transport systems (mechanical parameters).

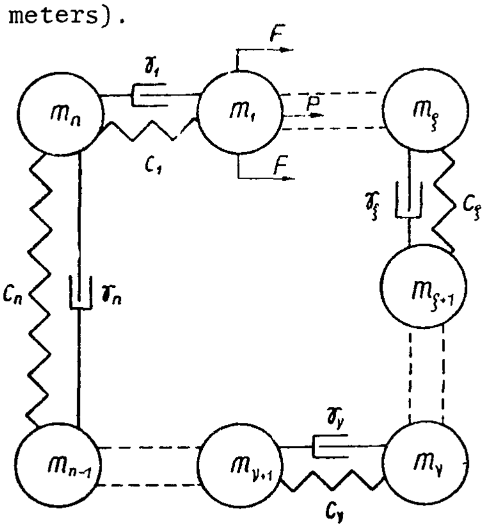

Figure 1. Dynamic Model. F, Friction force; P, Perturbing force; γ, Damping factor; C, Rigidity coefficient; m, Drive mass.

The statistical and dynamic characteristics of the systems studied, after composition and solution of mathematical models, cannot fully show the concrete mechanical and dynamic parameters of greatest significance for the value and nature of statistical and dynamic characteristics. Dynamic models of systems studied are composed, nonlinear differential equations with stochastic excitation are described, algorithms are composed and programs are written for their machine solution. /124

One such dynamic model for a tape drive mechanism with a closed loop of magnetic tape is shown on Figure 1. The masses of sectors of magnetic tape m, interconnected by elasticity C and damper γ, are perturbed by the tape drive units and other sources of excitation consisting of friction forces F and perturbation forces P, which generally are random and unstable in nature.

In order to simplify and apply the solutions to engineering practice, it was assumed that the perturbing factors are random and stable in nature, and have ergodic properties.

These dynamic models were described by differential equations of the following type:

$$a_i \ddot{y}_i + \sum_{k=1}^{n} b_{i_k} \dot{y}_k + \sum_{k=1}^{n} c_{i_k} y_k + \sum_{k=1}^{n} l_{i_k} \dot{y}_k^2 = P_i(t)$$

where i = 1,2,...,n, and n is the number of degrees of freedom.

In this equation, the nonlinear expression for drive friction was approximated by a parabolic expression $b_{i_k} \dot{y}_k + l_{i_k} \dot{y}_k^2$, a_i represents the driven masses, b_{i_k} are the summary coefficients of damping and dry friction, c_{i_k} are the coefficients of elasticity, l_{i_k} are the coefficients of dry friction, $y_i = y_i(t)$ are the displacements of the driven masses, $P_i(t)$ are the perturbing forces applied to certain of the rollers or the magnetic tape. The differential equation is solved for two cases: 1) when $P_i(t)$ is a deterministic function, fixed by a Fourier series; 2) when $P_i(t)$ is a random stable function. In the latter case, the correlation function of the stable process $P_1(t)$, the values of which were fixed by a table, was considered known.

The problem was stated of determining displacements y_i, when $P_i(t)$ is a deterministic function and determination of the statistical characteristics of y_i and the dynamic characteristics of the mechanical systems when $P_i(t)$ is a random stable process.

NEW METHODS OF STUDYING THE DYNAMICS OF IMPACT PROCESSES

B. V. Rudgal'vis (Kaunas)

This work presents an investigation of the dynamics of impact processes, as well as vibration and impact stability of precision electromechanical systems. /125

It is established that modern impact stands, in place of a rectangular or sinusoidal half wave pulse as indicated in the technical characteristics, actually produce a pulse which is a polyharmonic attenuating oscillating process with an envelope of strongly distorted form, with a duration of several tens of oscillations. Spectral analysis of these processes has shown that the energy of the impact is transmitted in several significant components, lying in a band of frequencies up to several KHz. The dependence of the significant frequency components on the conditions of reproduction of the impact processes is determined. It is established that an increase in acceleration upon realization of an impact process causes the significant frequency components to increase in frequency.

A method is developed for performing spectral analysis, new parameters are suggested for evaluation of the nature and effectiveness of the impact process, such as the spectral density, spectral energy, effective duration, effective spectral width, form coefficient of impact pulse, as well as amplitude and frequency of the significant components defined on the curve of the spectrum.

A new apparatus is developed for recording one-time wide band impact processes with a scanning rate allowing automatic discretization for the performance of operative spectral analysis by computer. The principle of electrostatic recording is used in combination with a drum driven type recording device, the information carrier of which is held in position by a pressure drop on each side. This device allows the information carrier (electrographic paper strip) to be driven at a speed of up to 100 m/sec and the recording of impact processes, the frequency spectrum of which reaches 50 KHz with a curve rise slope of not over 82°. The time required to develop an image of the process is about 1 min.

Furthermore, the reactions of precision electromechanical systems to impact pulses with various spectral parameters were measured using electronic mathematical analogs. The zones of their vibration and impact stability were determined, recommendations were made for optimization of systems as concerns vibration impact stability.

NEW METHODS OF STUDYING AND INCREASING THE DYNAMIC ACCURACY OF PRECISION JIG BORING MACHINES

K. P. Dzidolikas (Kaunas)

This work analyzes an original device for measurement of the oscillations of a spindle, the accuracy of measurement of which is independent of the geometric errors in the shape of the support. It contains the sensor made with two series-connected capacitances, one of which changes as the spindle oscillates. An oscillogram records only the oscillations of the spindle which influence the accuracy and quality of operation directly.

Using this device, a study was made of the oscillations of a range of spindles of precision jig boring machines with an optical coordinate reading system. The realizations produced were computer processed. This work presents characteristic correlation functions and spectral densities of the oscillating process both during idle and during operation. This allowed the predominant sources of oscillations of the spindle to be determined.

In constructing the dynamic model of the units of the tool for the entire tool as a whole, various methods are used to determine the rigidity characteristic of elements of the tool. The work describes an original method for determining circular compliance diagrams, allowing a continuous curve of circular rigidity to be produced clearly on the screen of the oscilloscope. The method is based on the use of centrifugal forces as the loading forces and the recording of deformations in the plane using two capacitive sensors connected with a converting apparatus. This method is used to study the contact and natural deformations of elements of the machine and construct a circular diagram of the compliance of the spindles of machines relative to the tables.

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